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THE PARTITION OF THE LOAD IN RIVETED JOINTS.*

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INTRODUCTION.

RIVETED joints occur in many types of construction, and it is therefore of considerable practical importance to determine the exact manner in which they act, in order that a rational basis may be given for their design. The subject has attracted the attention of many experimenters, their investigations being mainly directed to a determination of the resistance of joints to rupture and of the frictional resistance to slip.[1] Attempts have also been made to determine the tension in the body of a rivet due to its contraction on cooling,[2] and Frémont has made an exhaustive study of the actual process of riveting and its effects upon the strength of the joint.[3] None of these experiments, however, have indicated very clearly the action of a riveted joint under working loads; *i.e.*, before permanent deformations of the plates or rivets have occurred. Very few attempts have been made either ex-

* Communicated by the author.

[1] A short bibliography of the subject up to the year 1909 is given in a paper by A. N. Talbot and H. F. Moore—"Tests of Nickel-steel Riveted Joints." *University of Illinois Bulletin*, No. 49. See also Preuss, *Zeit. Ver. Deut. Ing.*, 1912, p. 404, and C. Bach and R. Baumann, *Ibid.*, p. 1890.

[2] R. Baumann.

[3] Étude Expérimentale du Rivetage, Paris, 1906.

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perimentally to determine or theoretically to estimate the partition of the load among the rivets under working conditions. It is usually assumed, in designing a riveted joint, that the load is equally divided among the rivets. That this cannot be true in joints as usually designed has been generally recognized by writers on bridge design,[4] and illustrated by experiments on the distortion of lath and india-rubber models.[5]

The writer's attention was first directed to the subject by the results of a few readings taken with a Howard gauge on the cover plates of a large riveted splice tested by Prof. H. M. Mackay, of McGill University, for the Quebec Bridge Commission. These appeared to indicate that the rivets in the end rows received by far the greater portion of the load, and suggested to the writer that the actual partition of the load in any joint could be determined by extensometer measurements on the cover plates of the joint. It was first necessary to determine whether or not the strains of the outer surfaces of the cover plates could be used to obtain the mean strains in the cover plates. This was shown to be the case by the experiment given in Part II, §2 of the present paper, and experiments were commenced upon a series of butt joints having a single line of rivets. While making these experiments it occurred to the writer that the results might be interpreted and the partition of load worked out theoretically by means of the Principle of Least Work. This proved to be so, and further experiments verified the theory in every particular.

The present paper is divided into two parts. Part I shows how, by means of the Principle of Least Work, a series of equations may be obtained, giving the load carried by each rivet in any form of riveted joint in terms of a quantity K, which, if the rivets are in shear, depends upon the manner in which work is stored in the rivets; or, if they act by frictional hold on the plates, depends upon the work stored in the parts of the plates thus held. Part II gives the results of a series of experiments upon different forms of joints having a single line of rivets and loaded in tension, which confirm and illustrate the theory and show how the quantity K may be determined.

[4] See, for example, Burr, "Elasticity and Resistance of the Materials of Engineering," p. 700; Johnson, Bryan and Turncaure, "Framed Structures, part iii; Shaper, Eiserne Brücken.

[5] C. R. Young, *Trans. Can. Soc. C. Eng.*, 1906, part ii, p. 257.

It is felt that much of the usefulness of the theory given depends upon whether the rivets in a joint are in shear or whether they act by holding the plates together by friction. The later experiments on the slip [6] of rivets seem to show that at working loads the latter is the case, and the results of the experiments described in the present paper seem to bear this out. The writer has in hand experiments which he hopes will give definite information on the subject.

NOTE.—In all that follows, a *row* of rivets means a number of rivets arranged in a straight line perpendicular to the axis of load; and a *line* of rivets means a number of rivets arranged in a straight line parallel to the axis of load.

PART I.—THEORETICAL.

§1. *General Method of Analysis.*

A riveted joint may be considered as a statically indeterminate system in which, under the action of external forces, work is stored in the plates and the rivets. It follows, from the Principle of Least Work, that the stresses will be distributed in the different parts of the system in such a way that the elastic work stored is a minimum. In any actual riveted joint the distribution of stress in the plates is more or less complicated,[7] but a close approximation to the forces taken by the rivets may be obtained by assuming the stress to be uniform between any two rows of rivets. The modifying effect of a variable distribution will be considered later (§3). The work stored in the rivets themselves, if they are in shear, or in the parts of the plates immediately surrounding the rivets, if the latter act by holding the plates together, may be taken as kX^2 where X is the load on the rivet and k is a constant, no assumption being made as to the manner in which this work is stored. On the above basis a series of equations may be obtained for any given joint, giving the load taken by each rivet in terms of k.

§2. *The Double-cover Butt Joint with a Single Line of Rivets.*

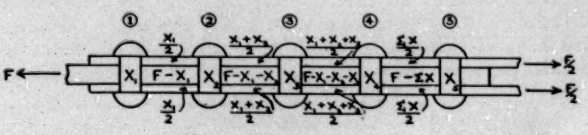
The simplest case is that of a double-cover butt joint having covers and plates of uniform width, the two cover plates having

[6] See Preuss, *loc. cit.*

[7] See a paper by Prof. E. G. Coker, *Trans. Inst. Naval Arch.* (England), 1913.

each the same area of cross-section, and the connection being made by a single line of rivets of uniform diameter and pitch (Fig. 1).

FIG. 1.



Let a_p represent the cross-sectional area of the middle plate,
a_c represent the cross-sectional area of each cover plate,
l represent the pitch of the rivets,
n represent the number of rivets on each side of the junction of the main plates,
F represent the load, tensile or compressive, carried by the joint, and X_1 represent the load carried by the 1st rivet, X_2, that carried by the second rivet, etc.

Then between the first and second rivets the load carried by the main plate is $F - X_1$, and the load carried by each cover plate is $\frac{X_1}{2}$; between the second and third rivets the load carried by the main plate is $F - X_1 - X_2$, and by each cover plate is $\frac{X_1+X_2}{2}$; and between the $(n-1)^{th}$ and n^{th} rivet the load carried by the main plate is $F - \sum_1^{n-1} X$ and by each cover plate is $\frac{\sum_1^{n-1} X}{2}$, where $\sum_1^{n-1} X = X_1 + X_2 + \ldots\ldots X_{n-1}$. The distribution of the load for five rivets is shown in Fig. 1, which represents one-half of the joint.

Now assuming that the stress in any portion of a plate between two rivets is uniformly distributed (see §1), the work stored in this portion, if the load carried by it is P, is $\frac{P^2 l}{2aE}$ where a is the cross-sectional area, l the pitch of the rivets,[8] and E is Young's Modulus for the material of the plate. It will be assumed that E is the same for both cover and middle plates.[9] The work stored in the rivets will be assumed to be of the form kX^2, as described above.

[8] l should really be taken a little less than the pitch of the rivets in order to allow for the portion of the plate cut away for the rivet hole.

[9] See § 3.

Then, if W represents the total work stored in one-half of the joint,

$$2EW = \frac{l}{a_p}\left[(F-X_1)^2+(F-X_1-X_2)^2+(F-X_1-X_2-X_3)^2+\ldots\ldots+(F-\Sigma X)^2\right]$$
$$+\frac{2l}{a_c}\left[\left(\frac{X_1}{2}\right)^2+\left(\frac{X_1+X_2}{2}\right)^2+\left(\frac{X_1+X_2+X_3}{2}\right)^2+\ldots\ldots+\left(\frac{\Sigma X}{2}\right)^2\right]$$
$$+k\left[X_1^2+X_2^2+X_3^2+\ldots\ldots+(F-\Sigma X)^2,\quad\right],\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

where $\Sigma X = X_1+X_2+X_3+\ldots\ldots+X_{n-1}$.

In accordance with the Principle of Least Work, the forces X_1, X_2, etc., in the above will take the values which will make W a minimum.

Thus

$$\frac{\partial W}{\partial X_1}=0,\ \frac{\partial W}{\partial X_2}=0,\ \frac{\partial W}{\partial X_3}=0\ \ldots\ldots\ \frac{\partial W}{\partial X_{n-1}}=0$$

But $$\frac{\partial W}{\partial X_1} = -\frac{2l}{a_p}\left[(F-X_1)+(F-X_1-X_2)+\ldots\ldots+(F-\Sigma X)\right]$$
$$+\frac{l}{a_c}\left[X_1+(X_1+X_2)+\ldots\ldots+\Sigma X\right]$$
$$+2k\left[X_1-(F-\Sigma X)\right].$$

Thus, equating this to zero and dividing through by $\frac{a_c}{l}$,

$$\left[(n-1)\left(1+\frac{2a_c}{a_p}\right)+\frac{4ka_c}{l}\right]X_1+\left[(n-2)\left(1+\frac{2a_c}{a_p}\right)+\frac{2ka_c}{l}\right]X_2$$
$$+\ldots\ldots+\left[\left(1+\frac{2a_c}{a_p}\right)+\frac{2ka_c}{l}\right]X_{n-1}=\left[(n-1)\frac{2a_c}{a_p}+\frac{2ka_c}{l}\right]F.$$

Writing $C=1+\frac{2a_c}{a_p}$ and $K=\frac{2ka_c}{l}$, and taking $F=1$ for convenience,

$$[(n-1)C+2K]X_1+[(n-2)C+K]X_2+\ldots\ldots+[C+K]X_{n-1}=(n-1)C+K$$

Differentiation with respect to X_2, X_3 etc., leads to the equations

$$[(n-2)C+K]X_1+[(n-2)C+2K]X_2+[(n-3)C+K]X_3+\ldots\ldots+[C+K]X_{n-1}$$
$$=(n-2)C+K,$$
$$[(n-3)C+K]X_1+[(n-3)C+K]X_2+[(n-3)C+2K]X_3+\ldots\ldots+[C+K]X_{n-1}$$
$$=(n-3)C+K,$$
$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$
$$[C+K]X_1+[C+K]X_2+[C+K]X_3+\ldots\ldots+[C+2K]X_{n-1}=C+K.$$

Thus a set of $(n-1)$ linear simultaneous equations has been established, from which $X_1, X_2 \ldots X_{n-1}$ may be found, while $X_n = 1 - \Sigma X$. For the sake of brevity in what follows, the above and all similar equations will be written in the abbreviated form

$$\begin{vmatrix} (n-1)C+2K & (n-2)C+K & (n-3)C+K & \ldots & C+K & (n-1)C+K \\ (n-2)C+K & (n-2)C+2K & (n-3)C+K & \ldots & C+K & (n-2)C+K \\ (n-3)C+K & (n-3)C+K & (n-3)C+2K & \ldots & C+K & (n-3)C+K \\ \ldots & \ldots & \ldots & \ldots & \ldots & \ldots \\ C+K & C+K & C+K & \ldots & C+2K & C+K \end{vmatrix} \quad (2)$$

It will be noticed that the quantities C and K in the above have no dimensions, but are simply numbers. C is determined from the ratio of thickness of cover plate to thickness of main plate. Thus, if each of the cover plates has half the thickness of the middle plate $C = 2$, if each cover plate is of the same thickness as the middle plate $C = 3$, etc. K must be determined by experiment in the absence of an exact theoretical estimate of the work stored in a rivet carrying a given load. This matter will be discussed later.

It would serve no useful purpose to give a general solution of the above equations for any number of rivets, as the equations may be solved easily for any particular case and the results are similar in form, whatever n may be. In the experiments described later the number of rivets on each side of the junction was usually five, so that the solution for $n = 5$ will be considered in detail.

In this case equations (2) reduce to

$$\begin{vmatrix} 4C+2K & 3C+K & 2C+K & C+K & 4(C-1)+K \\ 3C+K & 3C+2K & 2C+K & C+K & 3(C-1)+K \\ 2C+K & 2C+K & 2C+2K & C+K & 2(C-1)+K \\ C+K & C+K & C+K & C+2K & (C-1)+K \end{vmatrix} \quad \ldots\ldots(3)$$

Subtracting the second equation from the first, and the third from the second,

$$KX_3 = (C+K)X_1 - (C-1) \quad \ldots\ldots(4)$$

$$KX_4 = CX_1 + (C+1)X_2 - (C-1) \quad \ldots\ldots(5)$$

Eliminating X_4 from the third and fourth equations and substituting X_2 and X_3 from the above,

$$X_1 = \frac{K^4 + (C-1)10K^3 + (C^2-C)15K^2 + (C^3-C^2)7K + (C^4-C^3)}{5K^4 + 20CK^3 + 21C^2K^2 + 8C^3K + C^4} \quad \ldots\ldots(6)$$

This expression may be used to find X_1 for any particular values of C and K. X_2 and X_3 may then be determined from equations (4) and (5), X_4 from the last equation of (3), and $X_5 = 1 - \Sigma X$.

It may readily be seen from equation (6) that the assumption ordinarily made in designing riveted joints, *i.e.*, that all the rivets take an equal proportion of the load, would only be true if $K = \infty$, in which case $X_1 = X_2 = X_3 = X_4 = X_5 = \frac{1}{5}$. But this would mean that the rivets were quite flexible and offered no resistance to distortion, which would be a practical impossibility. Thus, in any joint of the type considered in this section, the load must be unequally distributed among the rivets.

If the rivets were absolutely rigid, *i.e.*, suffered no distortion when the joint was loaded, K would be equal to zero and

$$X_1 = \frac{C-1}{C},$$

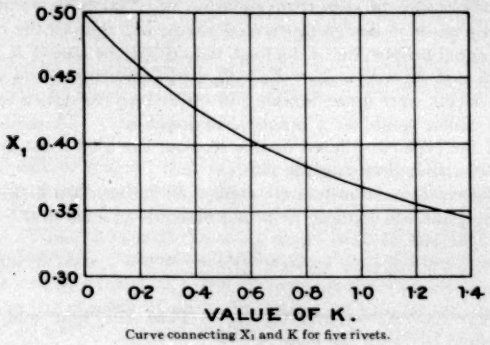
while $X_2 = X_3 = X_4 = 0$ and $X_5 = \frac{1}{C}$. Thus the first and last rivets would carry all the load.

If C were equal to 2, *i.e.*, if the cover plates were of the correct thickness, each one-half of the thickness of the main plate, these rivets would each take one-half of the load; if the covers were each equal in thickness to the main plate, the first rivet would carry two-thirds of the load and the last one-third; and if the covers were only one-quarter the thickness of the main plate, the first rivet would carry one-third of the load and the last two-thirds.

Actually the rivets are neither infinitely flexible nor infinitely rigid, but are elastic, and K has some finite value. Fig. 2 shows the value of X_1 for all values of K between 0 and 1.4 when there are five rivets. This curve will be used later in discussing the experimental results. The greater the value of K the more nearly uniform are the loads carried by the rivets, but, from the actual experimental values found from the specimens tested, it would appear that in most practical cases the two end rivets carry by far the greater part of the total load. The greatest experimental value of K was 1.30 for ½-inch rivets at about one and a half times the working load. At the working load it was approximately equal to unity. In order to simplify the present discussion this value will be taken in most of the illustrative cases.

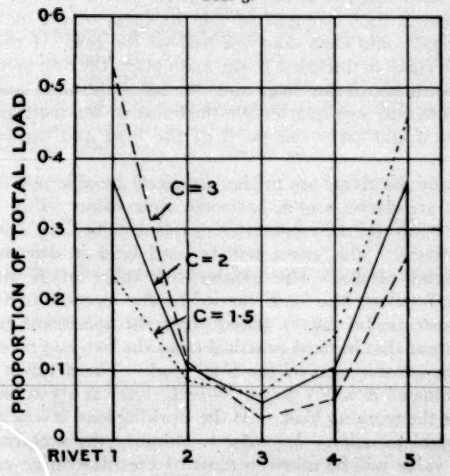
The proportion of the load carried by each rivet when $K = 1$ is tabulated in Table I for three values of C, while Fig. 3 shows the results graphically. These three examples will serve to show the

FIG. 2.



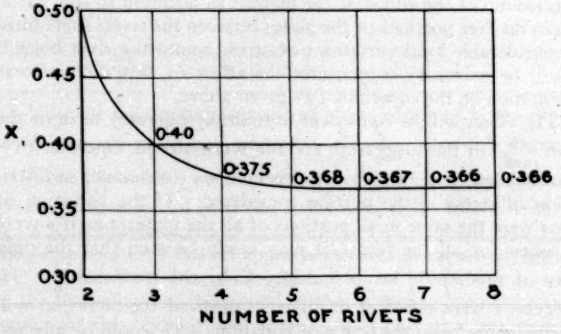
Curve connecting X_1 and K for five rivets.

FIG. 3.



general manner of distribution in most practical cases. In each case the end rivets take the greater part of the load, carrying 0.695, 0.736, and 0.793 of the total load when C = 1.5, 2, and 3.0 respectively. In no case does the middle rivet carry more than a very small fraction of the total load. It will be noticed that the distribution is symmetrical when the cover plates are of the correct thickness, *i.e.*, when $C = 2$. This is so no matter what value K may have, and, by taking account of this, the equations may be simplified by putting $X_2 = X_4$, thus reducing their number by one equation. When there is a large number of rivets, this shortens the solution considerably. For example, if $n = 10$, the number of

FIG. 4.



equations may be reduced from 9 to 6. When the thickness of each of the cover plates is more than one-half of the thickness of the main plate, the first rivet receives more than the last, as shown by the diagram for $C = 3$. On the other hand, when C is less than 2 this condition is reversed, the last rivet receiving more than the first.

Fig. 4 shows the proportion of the total load taken by the first rivet for $C = 2$, $K = 1$ for joints having 1–8 rivets on each side of the junction. It will be noticed that the curve becomes practically horizontal when the number 5 is reached, so that no matter how many more rivets may be added, the two end rivets

get practically the same proportion of the total load as with five rivets. This means that as the number of rivets increases each of the other rivets receives a less and less proportion of the load, and those near to the middle of the joint are practically idle.

§3. *Effect of Factors Neglected in the Above Analysis.*

(*a*) Effect of Non-uniform Distribution of Stress in the Plates.

It was assumed, in estimating the work stored in the plates, that the stress is distributed uniformly between any pair of rivets. A glance at Fig. 9, page 583, will show that this is not so, the stress varying from a minimum along the centre line of the rivets to a maximum at the edges of the plate. In addition to this variation in the free portions of the plates between the rivets there must be considerable local variations of stress round the rivet holes.[10] It will be necessary to consider the effect of this non-uniform distribution on the equations (2) given above.

The effect will be equivalent to multiplying every term of the form $\frac{P^2 l}{2aE}$ in the expression for the work stored, equation (1), by some coefficient α which will depend upon the manner of distribution of stress in the portion considered. If the variation of stress were the same in all portions of all the plates, *i.e.*, if α were the same for all the terms, it may easily be seen that the only effect of this would be to multiply K by the coefficient $\frac{1}{\alpha}$. If, however, α were different in different parts of the plates, as will usually be the case, the terms of equations (2) would be affected differently and there would be a modification of the distribution of load between the rivets. Fortunately the coefficient α cannot be very different from unity, as the following analysis will show.

Suppose that the stress in a portion of the cover plate between two rivets varies uniformly from f_1 at the centre line to f_2 at the edges. This is not far from the actual manner of distribution as shown in Fig. 9, page 583. Let the length of the portion considered be l, the breadth $2b$, and the thickness t, a_e being $2\,bt$. Then the mean stress in the plate is $\frac{f_1+f_2}{2}$. If this were uniformly

[10] This variation has been examined for a more or less analogous case by E. G. Coker, using his polarized light method. See *Trans. I. N. A.*, 1913.

distributed, as assumed in the analysis of §2, the work stored in the portion considered would be

$$W_1 = (f_1+f_2)^2 \frac{bdl}{4E}$$

The actual work stored, however, is

$$W_2 = 2\int_0^b \left\{f_1 + \frac{f_2-f_1}{b}\cdot x\right\}^2 \cdot \frac{dl}{2E}\cdot dx.$$

$$= \frac{bdl}{3E}\left\{f_1^2+f_1f_2+f_2^2\right\}$$

The ratio

$$r = \frac{W_2}{W_1} = \frac{4}{3}\cdot\frac{f_1^2+f_1f_2+f_2^2}{(f_1+f_2)^2} = \frac{4}{3}\left\{1-\frac{f_1f_2}{(f_1+f_2)^2}\right\}. \quad \ldots\ldots\ldots\ldots (7)$$

Now the maximum variation of stress observed in the experiments was about 20 per cent.; *i.e.*, $\frac{f_2}{f_1} = \frac{5}{4}$. For this variation $r = \frac{4}{3}\left\{1-\frac{20}{81}\right\} = 1.004$; *i.e.*, the work stored is about 0.4 per cent. greater than it would be if the stress were uniformly distributed. A longitudinal variation of stress of the same amount would give the same result. Thus no appreciable error is caused by neglecting this factor. The probably intense local variations of stress in the immediate neighborhood of the rivets are not easy to estimate, but they extend over a small area only and are probably similar in all the plates, so that they will be included in the experimental estimates of K, as will also be the error due to disregarding the parts of the plate cut away for the rivet holes in estimating the volume of the plate between a pair of rivets. Neither of these is likely to have any great effect upon the value of K.

(*b*) Effect of Unequal Partition of the Load Between the two Cover Plates.

It was assumed, in the analysis of §2, that the cover plates each received one-half of the load transmitted to them by the rivets. This was not so in most of the experiments described in Part II.

Suppose that one cover plate takes $\frac{1}{s}$ of the load, then the other takes $1-\frac{1}{s}$ and equation (1) becomes

$$2EW=\frac{l}{a_p}[(F-X_1)^2+\ldots\ldots]+\frac{l}{s^2a_c}[X_1^2+(X_1+X_2)^2+\ldots\ldots+(\Sigma X)^2]$$
$$+\frac{(s-1)^2}{s^2}[X_1^2+(X_1+X_2)^2+\ldots\ldots(\Sigma X)^2]+k\,[\,h_1^2+X_2^2+\ldots\ldots+(F-\Sigma X)^2]$$

TABLE I.

C	X_1	X_2	X_3	X_4	X_5
1½	0.235	0.086	0.068	0.151	0.460
2	0.368	0.105	0.054	0.105	0.368
3	0.528	0.112	0.034	0.061	0.265

On differentiating this with respect to X_1, X_2, etc., and forming the equations as before, it will be found that the equations take exactly the same form as in the last section if C be written for

$$\frac{2a_c}{a_p}+2\left(1-\frac{2}{s}+\frac{2}{s^2}\right)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

instead of for $\frac{2a_c}{a_p}+1$, as in equations (2).

Thus the effect of the non-equipartition of the load is equivalent to a change of C.

Suppose, for example, that $\frac{1}{s}=0.6$, the cover plates being of the correct thickness for which C would be equal to 2 if the load were equally divided. Then

$$C=2[0.5+1-1.2+0.72]$$
$$=2.04.$$

If $K=1$, the effect of this change of C may be seen from the annexed table. The load taken by the first rivet is increased about 2.44 per cent., and the load taken by the last rivet is decreased 1.36 per cent.

In practical cases, owing to want of straightness of the plates, etc., it will often happen that s is not the same throughout. This

could be allowed for in a similar manner, but, of course, no general rule can be given. The effect is not likely to be great in any case, as may be seen from the above. It should, however, be allowed for in reducing experimental data to obtain exact values of K.

TABLE II.

C	X_1	X_2	X_3	X_4	X_5
2.00	0.368	0.105	0.054	0.105	0.368
2.04	0.377	0.106	0.052	0.102	0.363

(*c*) Effect of the Main Plate and Cover Plates Having Different Moduli of Elasticity.

It often happens that the cover plates, being thinner than the main plates, have a greater modulus of elasticity. Let E_c be the modulus of the cover plates, E_p that of the main plate, and let the ratio $\frac{E_c}{E_p} = r$. It may be shown from equation (1) that the effect of r being different from unity is to change C from $1 + \frac{2a_c}{a_p}$ to

$$C = 1 + \frac{2a_c}{a_p} \cdot r \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

For example, if $E_c = 31 \times 10^6$ pounds per square inch and $E_r = 29 \times 10^6$ pounds per square inch, $r = \frac{31}{29}$, and, if the covers are each of half the thickness of the main plate,

$$C = 2.069.$$

The effect is thus similar to that discussed in (*b*).

Thus factors (*b*) and (*c*) both tend to increase the load taken by the first rivet and decrease that taken by the last.

§4. *Joint in Which the Middle Plate is of Variable Width, as in the Connection of Members to a Gusset Plate.*

In this section an analysis will be made of a joint of the type shown in Fig. 6*a*, page 573, which consists of two similar plates of uniform width attached one on each side of a plate of variable

width. In such a connection there will be no bending of the plates if the load carried by each of the outer plates is the same.

The analysis will be given for five rivets, but may readily be extended to any number. The same assumptions will be made as in §2.

Let a_e represent the cross-sectional areas of the outer plates, a_1, a_2, a_3 and a_4 . . the cross-sectional areas of the gusset plate mid-way between rivets 1 and 2, between rivets 2 and 3, etc., respectively. Then, assuming uniform distribution of stress in the plates between each pair of rivets, the work stored in each outer plate between any pair of rivets is $\frac{P^2 l}{2a_e E}$, as in §2, but the work stored in the middle plate will take a more complicated form because of the variable width. Consider the portion of the middle plate between rivets 1 and 2. Then, if the load on this portion is P,

$$2EW = \Sigma \frac{P^2 dx}{A}$$

$$= \frac{P^2}{t} \int_0^l \frac{dx}{b_2 + \frac{b_1 - b_2}{2} \cdot x}.$$

$$= \frac{P^2 l}{(b_1 - b_2)t} \cdot \log_e \frac{b_1}{b_2},$$

where b_1 and b_2 are the widths of the plate at rivet 1 and rivet 2 respectively and t is its thickness. If the plate were of uniform width, $\frac{b_1+b_2}{2}$,

$$2EW_1 = \frac{2P^2 l}{b_1 + b_2}$$

Thus the ratio

$$\eta_1 = \frac{W}{W_1} = \frac{b_1 + b_2}{2(b_1 - b_2)} \log_e \frac{b_1}{b_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

Thus the work stored in the portions of the middle plate may be expressed as $\eta_1 \frac{P_1^2 l}{2a_1 E}$, $\eta_2 \frac{P_2^2 l}{2a_2 E}$, etc., where η is a coefficient calculated from equation (10). On substituting these values in the work equation (equation (1), § 2), the first term becomes

$$l\left[\frac{\eta_1}{a_1}(F - X_1)^2 + \frac{\eta_2}{a_2}(F - X_1 - X_2)^2 + \ldots\ldots + \frac{\eta_4}{a_4}(F - \Sigma X)^2\right]$$

the other terms remaining as before. The method of §2 then leads to the equations

$$\begin{vmatrix} 4+2a_e\sum_1^4\frac{\eta}{a}+2K & 3+2a_e\sum_2^4\frac{\eta}{a}+K & 2+2a_e\sum_3^4\frac{\eta}{a}+K & 1+2a_e\sum_4^4\frac{\eta}{a}+K & 2a_e\sum_1^4\frac{\eta}{a}+K \\ 3+2a_e\sum_2^4\frac{\eta}{a}+K & 3+2a_e\sum_2^4\frac{\eta}{a}+2K & 2+2a_e\sum_3^4\frac{\eta}{a}+K & 1+2a_e\sum_4^4\frac{\eta}{a}+K & 2a_e\sum_2^4\frac{\eta}{a}+K \\ 2+2a_e\sum_3^4\frac{\eta}{a}+K & 2+2a_e\sum_3^4\frac{\eta}{a}+K & 2+2a_e\sum_3^4\frac{\eta}{a}+2K & 1+2a_e\sum_4^4\frac{\eta}{a}+K & 2a_e\sum_3^4\frac{\eta}{a}+K \\ 1+2a_e\sum_4^4\frac{\eta}{a}+K & 1+2a_e\sum_4^4\frac{\eta}{a}+K & 1+2a_e\sum_4^4\frac{\eta}{a}+K & 1+2a_e\sum_4^4\frac{\eta}{a}+2K & 2a_e\sum_4^4\frac{\eta}{a}+K \end{vmatrix}$$

where

$$\sum_1^4\frac{\eta}{a}=\frac{\eta_1}{a_1}+\frac{\eta_2}{a_2}+\frac{\eta_3}{a_3}+\frac{\eta_4}{a_4}$$

$$\sum_4^4\frac{\eta}{a}=\frac{\eta_4}{a_4}, \quad \text{etc.}$$

or writing

$$\alpha=4+2a_e\sum_1^4\frac{\eta}{a}+K,$$

$$\beta=3+2a_e\sum_2^4\frac{\eta}{a}+K,$$

$$\gamma=2+2a_e\sum_3^4\frac{\eta}{a}+K,$$

$$\delta=1+2a_e\sum_4^4\frac{\eta}{a}+K,$$

the equations become

$$\begin{vmatrix} \alpha+K & \beta & \gamma & \delta \\ \beta & \beta+K & \gamma & \delta \\ \gamma & \gamma & \gamma+K & \delta \\ \delta & \delta & \delta & \delta+K \end{vmatrix} \begin{vmatrix} \alpha-4 \\ \beta-3 \\ \gamma-2 \\ \delta-1 \end{vmatrix} \quad \ldots\ldots\ldots\ldots\ldots (11)$$

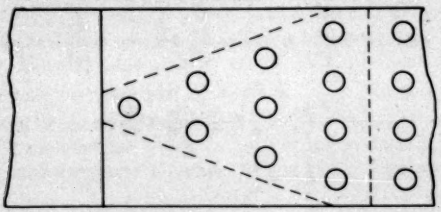
These equations are similar in form to those obtained in §2 and the same method of solution may be used. An example is given in Part II, §5.

§5. *Splices with Various Groupings of Rivets.*

The joints considered in the former sections contained a single line of rivets only. When a large number of rivets is required for a connection, it is usual to group the rivets in several rows, a different number in each row, as, for example, in the

splice shown in Fig. 5, which contains one rivet in the first row, two in the second, three in the third, etc. The distribution of the load between the various rivets of such a splice may be calculated by a similar method to that used in the cases considered above if the additional assumption is made that all the rivets in each row take the same load, which is probably nearly true in most

FIG. 5.



cases, although experimental data are not yet available on this point.

Let η_1 be the number of rivets in the first row, η_2 the number in the second row, etc., and, for simplicity, consider that there are only five rows on each side of the junction. Let X_1 be the total load taken by the first row of rivets, X_2 that taken by the second row, etc. Then each rivet of the first row will take the load $\frac{X_1}{n_1}$, each rivet of the second row $\frac{X_2}{n_2}$, etc. The work equation will be

$$2EW = \frac{l}{a_p}\left[(F-X_1)^2+(F-X_1-X_2)^2+\ldots\ldots+(F-\Sigma X)^2\right]$$
$$+\frac{2l}{a_c}\left[\left(\frac{X_1}{2}\right)^2+\left(\frac{X_1+X_2}{2}\right)^2+\ldots\ldots+\left(\frac{\Sigma X}{2}\right)^2\right]$$
$$+k\left[n_1\left(\frac{X_1}{n_1}\right)^2+n_2\left(\frac{X_2}{n_2}\right)^2+\ldots\ldots+n_5\left(\frac{F-\Sigma X}{n_5}\right)^2\right]\ldots\ldots\ldots(12)$$

This leads to the equations

$$\begin{vmatrix} 4C+K\left(\frac{1}{n_1}+\frac{1}{n_5}\right) & 3C+\frac{K}{n_5} & 2C+\frac{K}{n_5} & C+\frac{K}{n_5} & 4(C-1)+\frac{K}{n_5} \\ 3C+\frac{K}{n_5} & 3C+K\left(\frac{1}{n_2}+\frac{1}{n_5}\right) & 2C+\frac{K}{n_5} & C+\frac{K}{n_5} & 3(C-1)+\frac{K}{n_5} \\ 2C+\frac{K}{n_5} & 2C+\frac{K}{n_5} & 2C+K\left(\frac{1}{n_3}+\frac{1}{n_5}\right) & C+\frac{K}{n_5} & 2(C-1)+\frac{K}{n_5} \\ C+\frac{K}{n_5} & C+\frac{K}{n_5} & C+\frac{K}{n_5} & C+K\left(\frac{1}{n_4}+\frac{1}{n_5}\right) & (C-1)+\frac{K}{n_5} \end{vmatrix} \quad (13)$$

C and K having the same meanings as in §2. The method of solution is the same as for the equations of the preceding sections. Similar equations for any number of rows of rivets may be built up from a consideration of the above.

It will be interesting to consider one or two numerical examples in order to illustrate the method and to see how far the ordinary assumption of design—that each rivet carries an equal proportion of the load,—is justified. Take first the splice shown in Fig. 5, having four rows of rivets containing 1, 2, 3 and 4 rivets respectively. Let $C = 2, K = 1$ and $F = 1$. Equations (13) reduce to

$$\left|\begin{array}{c|c|c||c} 7.25 & 4.25 & 2.25 & 3.25 \\ 4.25 & 4.75 & 2.25 & 2.25 \\ 2.25 & 2.25 & 2.583 & 1.25 \end{array}\right|$$

TABLE III.

Specimen	Diameter of rivets	Thickness of middle plate	Thickness of each outer plate
A	$\frac{1}{2}''$	$\frac{3}{8}''$	$\frac{5}{16}''$
B	$\frac{3}{4}''$	$\frac{1}{2}''$	$\frac{1}{4}''$
C	$\frac{7}{8}''$	$\frac{1}{2}''$	$\frac{1}{4}''$
D	$\frac{3}{4}''$	$\frac{1}{4}''$	$\frac{3}{8}''$
E	$\frac{3}{4}''$	$\frac{1}{4}''$	$\frac{1}{4}''$
F and F'	$\frac{3}{4}''$	$\frac{1}{4}''$	$\frac{1}{4}''$

These give

$$X_1 = 0.339,\ X_2 = 0.034,\ X_3 = 0.159,\ \text{and}\ X_4 = 0.468.$$

These are the total loads carried by each row. Thus

each rivet in the first row takes 0.339,
each rivet in the second row takes 0.017,
each rivet in the third row takes 0.053,
each rivet in the fourth row takes 0.117.

Thus more than one-third of the total force is carried by the first rivet, while the middle rows take very little load. The load carried by the first rivet is so great that it will probably fail. If

this happens, or if the first rivet is removed, the values of X will become

$$X_2=0.416,\quad X_3=0.121,\quad X_4=0.463.$$

and

each rivet of the second row will take 0.208,
each rivet of the third row will take 0.040,
each rivet of the fourth row will take 0.116.

Thus the distribution is somewhat improved, but if the splice is in tension, the main plate is weakened by two rivet holes.

If the second row of rivets be removed,

$$X_3=0.441,\quad X_4=0.559.$$

Thus

each rivet in the third row takes 0.147,
each rivet in the fourth row takes 0.140,

and the distribution is much more uniform.

Usually in a splice of the type considered the cover plates are cut away as shown by the dotted lines in Fig. 5. This will alter the distribution of stress. Suppose, for example, that the widths of the cover plates at the first, second, third, and fourth rows of rivets are in the ratio 1 : 2 : 3 : 4. The equations will take the form

$$\begin{vmatrix} 3+a_c\frac{3}{1}\frac{\eta}{a}+K\left(\frac{1}{n_1}+\frac{1}{n_4}\right) & 2+a_c\frac{3}{2}\frac{\eta}{a}+\frac{K}{n_4} & 1+a_c\frac{3}{3}\frac{\eta}{a}+\frac{K}{n_4} & 3+\frac{K}{n_4} \\ 2+a_c\frac{3}{2}\frac{\eta}{a}+\frac{K}{n_4} & 2+a_c\frac{3}{2}\frac{\eta}{a}+K\left(\frac{1}{n_2}+\frac{1}{n_4}\right) & 1+a_c\frac{3}{3}\frac{\eta}{a}+\frac{K}{n_4} & 2+\frac{K}{n_4} \\ 1+a_c\frac{3}{3}\frac{\eta}{a}+\frac{K}{n_4} & 1+a_c\frac{3}{3}\frac{\eta}{a}+\frac{K}{n_4} & 1+a_c\frac{3}{3}\frac{\eta}{a}+K\left(\frac{1}{n_3}+\frac{1}{n_4}\right) & 1+\frac{K}{n_4} \end{vmatrix} \tag{14}$$

where a_c is the area of cross-section of the cover plates when of the same width as the middle plate.

Substituting numerical values.

$$\eta_1=\tfrac{1}{2}\log_e 2=1.0397,$$
$$\eta_2=\tfrac{1}{2}\log_e 1.5=1.0136,$$
$$\eta_3=\tfrac{1}{2}\log_e \tfrac{4}{3}=1.007,$$

and the equations become

$$\begin{vmatrix} 9.795 & 5.022 & 2.401 & 3.25 \\ 5.022 & 5.522 & 2.401 & 2.25 \\ 2.401 & 2.401 & 2.734 & 1.25 \end{vmatrix}$$

From these

$$X_1 = 0.225,\ X_2 = 0.145,\ X_3 = 0.132,\ X_4 = 0.498.$$

Thus

each rivet in the first row takes 0.225,
each rivet in the second row takes 0.072,
each rivet in the third row takes 0.044,
each rivet in the fourth row takes 0.125.

Thus shaping the cover plate decreases the load taken by the first rivet. If the first rivet be omitted

$$X_2 = 0.338,\ X_3 = 0.160,\ X_4 = 0.502$$

and

each rivet in the second row takes 0.169,
each rivet in the third row takes 0.053,
each rivet in the fourth row takes 0.125.

If the second row be also omitted

$$X_3 = 0.458,\ X_4 = 0.542$$

and

each rivet of the third row takes 0.153,
each rivet of the third row takes 0.135.

These illustrations are sufficient to show how the method may be used to determine the partition of load in any form of splice. The problem of the best arrangements of rivets in splices will be deferred until further experiments have been made.

§6. *Joints Having Rivets of Different Sizes or for Which the Values of K are Different.*

Consider a joint having a single line of rivets, five on each side of the junction. Let the values of K for the rivets be K_1, K_2, K_3, K_4 and K_5 for the first, second, third, fourth, and fifth rivets respectively. Then the work equation will be similar to that given in §2, equation (1), but the last term will be

$$K_1X_1^2 + K_2X_2^2 + \ldots + K_5(F - \Sigma X)^2.$$

Thus the equations for this case will be

$$\begin{vmatrix} 4C+K_{12}+K_5 & 3C+K_5 & 2C+K_5 & C+K_5 & 4(C-1)+K_5 \\ 3C+K_5 & 3C+K_2+K_5 & 2C+K_5 & C+K_5 & 3(C-1)+K_5 \\ 2C+K_5 & 2C+K_5 & 2C+K_3+K_5 & C+K_5 & 2(C-1)+K_5 \\ C+K_5 & C+K_5 & C+K_5 & C+K_4+K_5 & (C-1)+K_5 \end{vmatrix} \quad \ldots\ (15)$$

§7. *Lap Joints.*

In lap joints the loads on the two plates are not in the same straight line. This causes bending of the plates, and the distribution of stress may be considerably modified by this action. The rivets are in single shear, and this will affect the value of K. In view of these factors and in the absence of experimental data, no attempt will be made in the present contribution to give a complete theory of such joints. If the bending of the plates is neglected, the work equation for a lap joint having a single line of rivets and connecting two similar plates will be

$$2EW = \frac{l}{a_p}[(F-X_1)^2+(F-X_1-X_2)^2+\ldots\ldots+(F-\Sigma X)^2]$$
$$+\frac{l}{a_p}[X_1^2+(X_1+X_2)^2+\ldots\ldots+(\Sigma X)^2]$$
$$+k\,[X_1^2+X_2^2+\ldots\ldots+(F-\Sigma X)^2]$$

where a_p is the cross-sectional area of the plates.

Differentiating with respect to X_1, and equating the result to zero,

$$-\frac{2l}{a_p}[(F-X_1)+(F-X_1-X_2)+\ldots\ldots+(F-\Sigma X)]$$
$$+\frac{2l}{a_p}[X_1+(X_1+X_2)+\ldots\ldots+(\Sigma X)]$$
$$+2k\,[X_1-(F-\Sigma X)]=0$$

Thus, if $K=\frac{k\cdot a_p}{l}$ the equations are

$$\begin{vmatrix} (n-1)+2K & (n-2)+K & (n-3)+K & \ldots & 1+K & (n-1)+K \\ (n-2)+K & (n-2)+2K & (n-3)+K & \ldots & 1+K & (n-2)+K \\ (n-3)+K & (n-3)+K & (n-3)+2K & \ldots & 1+K & (n-3)+K \\ \ldots\ldots & \ldots\ldots & \ldots\ldots & \ldots & \ldots\ldots & \ldots\ldots \\ 1+K & 1+K & 1+K & \ldots & 1+2K & 1+K \end{vmatrix} \quad (16)$$

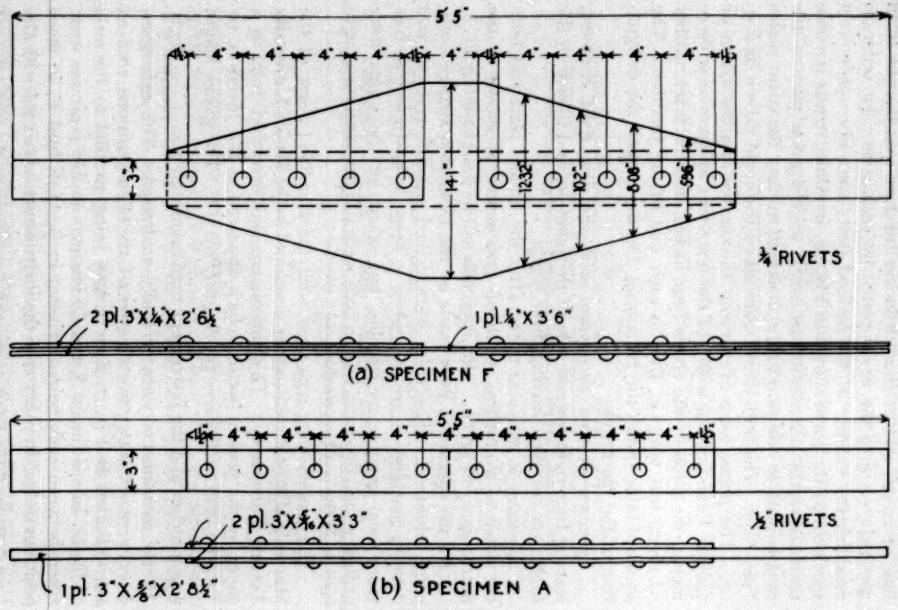
Thus the distribution of load is the same as for a butt joint having $C=2$, as considered in §2, if the value of K is the same. The above equations must be considered only as a first approximation.

PART II.—EXPERIMENTAL.

§1. *Specimens Used and Method of Experiment.*

The experiments described in the following pages were made with the object of determining the distribution of stress in the cover plates of a series of riveted butt joints having a single

FIG. 6.



Typical specimens.

line of rivets, and of thus deducing the load transferred from the main plate to the cover plates by each rivet. In order to establish the validity of the method, tests were also made on a specimen of the form shown in Fig. 6*a*, in which it was possible to measure the distribution of stress over a great part of the middle plate in addition to the distribution over the cover plates (see § 2). The butt joints were all of the form shown in Fig. 6*b*. The plates were of varying thicknesses and the rivets of various sizes, but the width of the plate was in every case three inches, the pitch of the rivets four inches, and the total number of rivets was ten; *i.e.*, five on each side of the junction. The annexed table shows the remaining dimensions and the designation of the specimens.

Specimen A was also tested with the first rivet removed, leaving four rivets on one side of the junction, and then with the fifth rivet removed, leaving three rivets.

Specimen F was also tested after the middle plate had been cut down to a uniform width of 4.09 inches, as shown by the dotted lines in Fig. 6*a*. This specimen will be designated by F^1.

All the specimens were made by the Dominion Bridge Company, Montreal. The holes were drilled and the riveting was done by machine. Care was taken to keep the specimens as free from local bending as possible, but otherwise the joints were ordinary shop products, and as such were subject to minor irregularities in the position of the holes, etc.

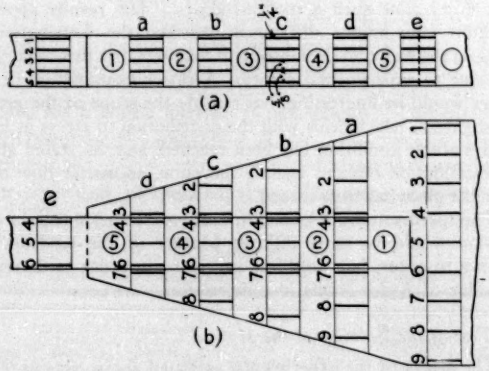
The strains were measured with the simplified form of the Marten's Mirror Extensometer developed in the McGill University Testing Laboratory. This instrument was described fully in a paper by the present writer which appeared in the JOURNAL OF THE FRANKLIN INSTITUTE, August, 1915.[11] The gauge length was 2 inches and the instrument read accurately to $\frac{1}{100,000}$ inch. All the instruments were carefully calibrated. The extensometers were set between each pair of rivets in positions such as are indicated in Fig. 7, their length being parallel to the axis of load. In specimens A and C, readings in five positions were taken between each pair of rivets; in B, D, E and F, in three positions only. In every case the instruments were read with the

[11] C. Batho, "The Effect of the End Connections on the Distribution of Stress in Certain Tension Members," J. F. I., August, 1915, p. 129.

knife-edge at each end in turn and the mean of these readings taken in order to eliminate errors due to bending. The strains measured in this way may be regarded as proportional to the stresses, since the strains perpendicular to the axis of load are so small as to be negligible. Readings taken on the middle plate below the joint and on the cover plates at the central section enabled the value of the modulus of elasticity for each plate to be determined.

The specimens were loaded in the 150,000-pound Emery testing machine at McGill University. This machine is of the

FIG. 7.



Positions of Extensometers.

vertical type and is very suitable for extensometer work because of the entire absence of vibration. The tests were carried out in a uniform manner. Four, or in some cases two, extensometers being placed in position and the mirrors set under an initial load of 100 pounds, the maximum load was applied and removed several times, the mirrors reset if necessary, and the load was then run up gradually, readings being taken at the required loads. The load was then reduced to its initial value. In nearly all cases the extensometer readings returned to zero. If not, the process was repeated until they returned satisfactorily. This latter precaution was, however, seldom necessary.

It will be seen that the success of the method depends upon the distribution of stress remaining the same after many loadings. This point was very carefully tested by repeating readings at intervals. No differences were found that were not within the range of experimental error; *i.e.*, the readings checked to $\frac{1}{100,000}$ inch. An attempt was made in specimen A to obtain readings on the first loading of the piece. This was found rather difficult as the ends of the specimen always slip a little in the grips when the load is applied for the first time, disturbing the extensometers, so that the loading has to be carried out in stages, returning to the initial load and resetting the extensometers every time such a motion occurs. The results appeared to show some minor differences between the distribution of stress on the first and on subsequent loadings, but these may have been due to experimental errors. A closer examination of this matter would be interesting, but outside the scope of the present investigation, which deals with the distribution of stress in joints when a stable condition has been reached, and, as stated above, this distribution remains exactly the same, no matter how many times the piece has been loaded.

The specimens were loaded in tension. Experiments in compression would be more difficult because of the tendency for bending to occur, but would be necessary in order to determine the value of K for joints in compression.

§2. *Test of the Validity of the Method.*

The object of the experiments, as stated above, was to obtain the proportion of the load transmitted from the middle plate to the cover plates by each rivet. Since extensometer measurements could be taken only on the outer surfaces of the cover plates, it will be seen that the validity of the method depends upon whether or no the strains in the cover plates deduced from these measurements were a true estimate of the mean strains in the plates. If, for example, the plates were held together mainly by friction between the plates, the stresses at the inner surfaces of the plates would probably be much greater than at the outer. In order to obtain information on this point, experiments were made on specimen F, in which the middle plate was much wider than the outer plates, so that measurements could be taken both

on the cover plates and on the middle plate. The extensometer positions are shown in Fig. 7, and Table IV gives the sum of the four extensometer readings on the two sides of the specimen at each position, one-hundred-thousandth of an inch being taken

TABLE IV.

Section	Sum of the extensometer readings on the two faces of the specimen								
	1	2	3	4	5	6	7	8	9
Central section of middle plate	76	100	113	167	179	162	130	101	70
a	103	110	112	76	67	77	126	118	107
b		122	107	110	90	109	117	111	
c		119	121	140	114	142	124	126	
d			123	162	150	166	128		
e				279	281	281			

as unit, for a tensile load of 16,000 pounds. The extensions were measured over a length of 2 inches. Thus, since the readings are correct to $\frac{1}{100,000}$ inch, an error of about 150 pounds, *i.e.*, about 1 per cent. of the load, is possible in the estimate of the stress. The error of the sum is probably much less than this.

TABLE V.

$E = 28.4 \times 10^6$ pounds per square inch for middle plate,
$= 30.4 \times 10^6$ pounds per square inch for outer plates.

Section	Mean strain in middle plate $\times 4$ $\left(\frac{1}{100,000}\right)''$	Mean strain in outer plate $\times 4$ $\left(\frac{1}{100,000}\right)''$	Load carried by middle plate Lbs.	Load carried by outer plates Lbs.	Total load from extensometer reading Lbs.	Error
						Per cent.
Central	127.7	.....	16000	.....	16000	.....
a	112.7	71.7	12320	4100	16420	+2.62
b	114.2	99.7	10340	5690	16030	+0.19
c	122.5	127.5	8800	7260	16060	+0.38
d	125.5	157.0	6650	8940	15590	−2.56
e	.....	280.5	.....	16000	16000	.....

The values of E for the plates were determined from the readings at the central section for the middle plate, and at section *e*, below the rivets, for the outer plates. The results are given at the head of Table V.

Columns 1 and 2 in Table V give the mean strains at each

section determined from the figures in Table IV. To obtain the mean strains in the outer plates, twice the middle reading was added to the two outer readings and the result divided by four. Thus at section *a* the mean strain

$$71.7 = \frac{76 + 2 \times 67 + 77}{4}.$$

The reason for this will be explained in §3. The mean strains in the inner plate were obtained by taking the average of the figures given in Table IV. The loads carried by the outer and inner plates respectively were calculated by using the values of E obtained as explained above and are given in columns 3 and 4 of Table V.

At each section the sum of these, if the estimated strains are the mean strains in the plates, should be 16,000 pounds, the total load. The actual sums are given in column 5, and it will be noticed how close they are to this value. The percentage errors are given in column 6. They are quite negligible at sections *b* and *c* and only amount to about 2.6 per cent. at sections *a* and *d*. Thus the validity of the method may be regarded as fully established. The agreement of the results will be seen to be remarkably close when it is considered that the experimental error may amount to about 1 per cent. and that readings could not be taken over the parts of the middle plates covered by the outer plates.

It may occur to the reader that there is also an error due to the varying width of the middle plate not being taken into account in reducing the strains to stresses. This error, however, was estimated and found to be too small to be considered.

To sum up, the above experiment shows that extensometer measurements on the outside of the outer plates of a riveted joint are sufficient for the determination of the proportion of the load carried by each rivet, since they give accurate values for the mean strains in the plates. It also appears to prove that friction between the plates cannot play any part in transferring the load, except possibly in the parts of the plates close to the rivets.

§3. *Experimental Results for Specimens A, B and C in Which the Thickness of Each of the Cover Plates was One-half the Thickness of the Middle Plate.*

The results of the tests on specimens A, B, and C are given in Table VI, together with their reduction to find the load taken by each rivet. It has not been considered necessary to tabulate each individual extensometer reading. The figures in Table VI are the sums of the four readings at the corresponding positions on the two cover plates, the sum being given, for convenience, instead of the mean. Readings were also taken on the middle plates below the joint for the purpose of determining E for these plates, but are not tabulated.

The earlier experiments were made on specimens B, D, E and F, and in these readings were taken in three positions only at each cross-section, corresponding to 1, 3 and 5, Fig. 7. The results (see Table VI) showed clearly that the distribution of strain across the width of the plates was not uniform, the readings at the central position being always less than at the outer positions. It was therefore thought to be advisable to take readings at the intermediate positions, 2 and 4, Fig. 7, in order to study more closely the actual distribution of strain. This was done in specimens A and C.

The results for these specimens given in Table VI show that the distribution of strain is of substantially the same character at all loads and for each specimen. The strain is always least at the centre, rising to a maximum at each outer boundary. The readings for specimen A at the cross-section, between the fourth and fifth rivets, where the variations are most marked, are plotted in Fig. 8. It will be noticed that the shape of the curves is the same at all loads. The ratio of the mean of the readings at 1 and 5 to the reading at 3 has the values

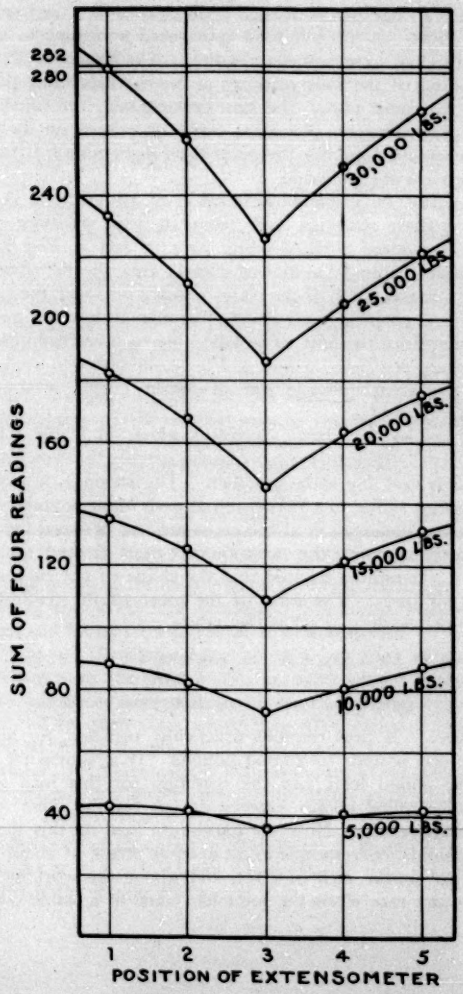
1.15, 1.20, 1.22, 1.23, 1.22, 1.21

at the loads

5000, 10000, 15000, 20000, 25000, 30000 pounds

respectively. It thus remains practically constant for all loads from 10,000 pounds to 30,000 pounds. It is somewhat less at the 5,000 pound load, but the readings for this load are too small to be relied upon. Curves drawn for the other sections give similar results. Now the maximum load on this specimen, 30,000 pounds, corresponds to an average stress of about 15,280 pounds per square inch of rivet, well above the working stress. Thus, at any rate when the joint has come to a stable condition

FIG. 8.



Distribution of strain at different loads between the fourth and fifth rivets of specimen A.

TABLE VI

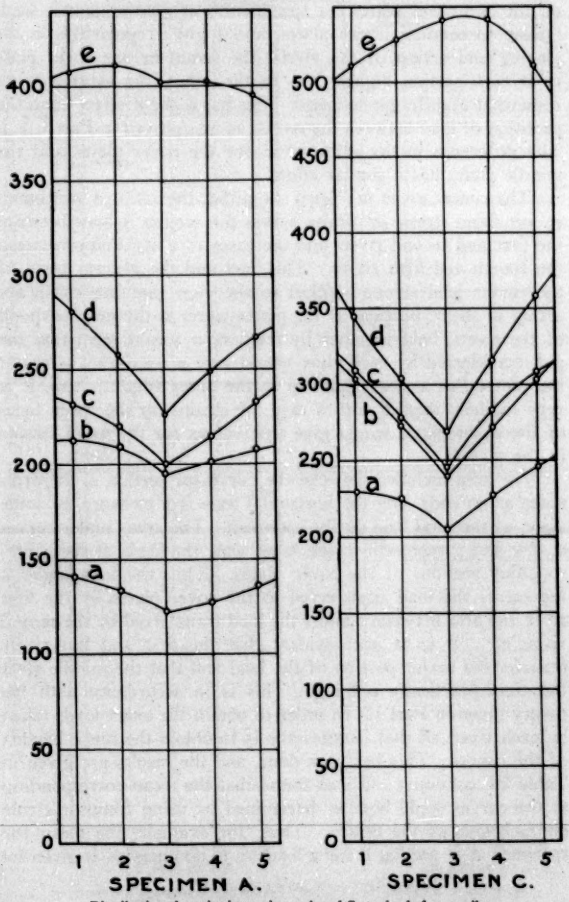
1	2																				3					4					5	6
Load	Sum of the four extensometer readings in corresponding positions on the two faces of the specimen																				Mean of the readings at each section (obtained as described on page 577)					Percentage of the total load taken by each rivet					Mean of 1 and 5.	K
	a					b					c					d					e												
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5	a	b	c	d	e	1	2	3	4	5		
A . E for middle plates = 30.4 × 10⁶ lbs. per sq. in., E for cover plates = 31.7 × 10⁶ lbs. per sq. in.																																					
5000	29	28	28	28	31	36	36	33	35	36	37	36	34	36	38	42	41	36	40	41	66	68	67	63	61	28	35	36	40	65	43.2	10.8	1.5	6.2	38.3	40.7	.....
10000	60	60	55	57	61	73	70	65	69	73	75	72	66	70	75	88	82	73	80	87	134	137	134	135	126	58	69	70	80	133	43.6	8.3	0.8	7.5	39.9	41.7	0.485
15000	85	83	77	80	85	108	107	102	114	111	113	109	99	108	115	135	125	109	121	131	200	202	200	196	192	81	108	108	122	198	40.9	13.6	0.0	7.1	38.4	39.6	0.673
20000	105	102	95	100	103	147	142	133	142	145	151	148	131	145	154	182	168	145	163	175	272	274	266	263	258	100	141	147	164	267	37.5	15.4	2.2	6.4	38.6	38.	0.837
25000	124	120	112	113	116	181	175	162	170	176	192	185	163	180	194	233	211	186	205	221	343	344	336	337	329	116	172	181	207	328	34.4	16.5	2.7	7.7	38.8	36.6	1.027
30000	141	134	123	127	136	214	209	195	201	208	232	222	200	214	233	281	258	226	249	267	408	411	402	402	395	131	204	217	252	404	32.4	18.0	3.2	8.7	37.6	35.0	1.300
A with 5th rivet removed.																																					
5000	36	32	32	32	35	40	37	36	35	38	45	39	37	39	43	.....					.....					33	36	40	.....	.....	50.8	4.6	6.1	38.5	....	44.6	0.273
10000	62	60	59	59	62	78	71	67	69	73	97	80	73	78	94	.....					.....					60	70	81	.....	.....	45.0	7.5	8.3	39.1	....	42.0	0.475
15000	89	83	83	86	94	115	105	104	105	111	141	122	111	120	138	.....					.....					86	106	122	.....	.....	43.5	10.1	8.1	38.4	....	40.9	0.573
20000	109	104	101	104	109	151	142	137	140	149	191	166	155	163	184	.....					.....					104	142	167	.....	.....	39.0	14.2	9.3	37.5	....	38.2	0.900
25000	126	119	120	125	129	190	177	172	174	184	241	209	195	205	235	.....					.....					123	177	210	.....	.....	36.4	16.0	9.8	37.9	....	37.1	1.066
30000	143	135	132	143	153	228	212	208	211	220	294	257	239	252	281	.....					.....					140	213	257	.....	.....	34.6	18.0	10.0	36.4	....	35.5	1.380
A with 1st and 5th rivets removed.																																					
5000	.....					37	34	34	35	35	45	39	36	39	42	.....					.....					.....	35	39	.....	.....	....	53.9	6.1	40.0	....	47.0	0.150
10000	.....					64	62	62	66	65	96	83	69	79	90	.....					.....					.....	63	81	.....	.....	....	47.4	13.5	39.1	....	43.3	0.450
15000	.....					95	92	91	92	92	139	127	105	120	135	.....					.....					.....	92	122	.....	.....	....	40.5	15.1	38.4	....	42.5	0.545
20000	.....					116	112	112	119	119	184	168	143	166	177	.....					.....					.....	115	165	.....	.....	....	43.1	18.7	38.2	....	40.7	0.825
25000	.....					143	141	137	143	140	238	211	183	208	221	.....					.....					.....	141	208	.....	.....	....	41.8	19.8	38.5	....	40.1	0.975
30000	.....					164	160	165	169	164	286	259	229	251	269	.....					.....					.....	164	254	.....	.....	....	40.6	22.3	37.2	....	38.9	1.320
C . E for middle plate = 29.4 × 10⁶ lbs. per sq. in., E for cover-plates = 30.4 × 10⁶ lbs. per sq. in.																																					
16000	135	129	119	131	148	161	146	127	147	161	161	149	132	152	161	175	155	135	156	182	264	282	299	292	265	130	146	149	156	280	46.5	5.7	1.1	2.5	44.3	45.4	0.225
20000	167	160	148	163	180	204	180	160	182	203	202	180	164	190	204	220	197	168	199	230	335	347	371	358	340	161	181	187	198	350	46.0	5.7	1.7	3.1	43.5	44.7	0.269
25000	199	192	180	201	215	253	227	200	229	254	255	232	205	235	259	278	246	215	250	288	420	439	457	450	423	195	227	232	248	438	44.5	7.3	1.1	3.0	43.5	44.0	0.314
30000	229	224	204	227	246	304	273	230	272	304	309	278	245	284	308	345	304	269	316	359	511	529	542	542	511	223	272	280	310	527	42.4	9.3	1.5	5.7	41.2	41.8	0.477
35000	258	249	235	261	280	354	319	277	319	356	360	326	285	335	365	406	361	320	368	425	595	617	629	635	605	254	318	328	365	616	41.2	10.4	1.6	6.0	40.7	41.0	0.545
B . E for middle plate = 30.1 × 10⁶ lbs. per sq. in., E for cover plates = 31.3 × 10⁶ lbs. per sq. in.																																					
16000	142	...	128	...	146	151	...	136	...	151	161	...	139	...	163	188	...	147	...	193	271	...	277	...	280	136	143	150	169	276	49.3	2.5	2.5	6.9	38.8	44.1	0.306
20000	173	...	153	...	179	199	...	169	...	200	199	...	167	...	197	234	...	184	...	240	343	...	347	...	345	164	184	182	212	345	47.6	5.8	−0.6	8.7	38.6	43.1	0.369
22000	189	...	169	...	194	217	...	182	...	222	219	...	182	...	219	265	...	204	...	267	378	...	382	...	379	180	201	200	237	380	47.5	5.5	−0.3	9.7	37.6	42.5	0.423
27000	201	...	205	...	247	244	...	220	...	279	257	...	223	...	280	322	...	250	...	337	462	...	468	...	464	214	241	246	290	465	46.0	5.8	1.1	9.4	37.7	41.8	0.477
30000	221	...	224	...	276	269	...	243	...	313	287	...	253	...	310	365	...	277	...	380	514	...	522	...	519	236	267	276	325	518	45.6	5.9	1.7	9.5	37.3	41.5	0.500
33000	241	...	244	...	295	299	...	269	...	344	320	...	273	...	344	406	...	307	...	425	563	...	573	...	572	256	295	302	361	569	45.0	6.9	1.2	10.3	36.6	40.8	0.564

after a few loadings, the manner of distribution of stress in the plates remains the same at all loads. This does not mean that the rivets carry the same proportions of the total load, but that there is no marked change in the way in which the load is transferred to the plates by the rivets.

It would require too much space to give all the curves for specimens A and C, and it is, fortunately, unnecessary, since they are, as shown above, similar for the same sections at different loads. Fig. 9 shows the readings at all the sections of the cover plates in specimens A and C for a load of 30,000 pounds. It will be remembered that the rivets in specimen A are ½ inch diameter and in specimen C are ⅞ inch diameter, the thickness of the cover plates being ¼ inch in specimen A and 5/16 inch in specimen C. In order to make direct comparison possible the ordinates of the curves for specimen A have been drawn to 5/4 the scale of the ordinates for specimen C, thus allowing for the difference in thickness of the plates. The curve at the central section, e, is higher in specimen C than in A, indicating that the value of E is lower in specimen C than in A. The actual values are 31.7×10^6 pounds per square inch for A and 30.4×10^6 pounds per square inch for C. Thus the ordinates do not represent the *stresses* in the two specimens to quite the same scale. The curves for sections a, b, c and d are very similar in form in the two specimens, although the load, 30,000 pounds, corresponds to an average load of about 15,280 pounds per square inch of nominal rivet section in A and of only 15,280 pounds per square inch in C, showing that the manner of distribution of stress at a cross-section is substantially the same not only at different loads, as shown above, but also in different specimens. All the experimental results show this in an equally striking manner. The curves are not quite symmetrical about the centre line. This is probably due to slight irregularities in the construction of the specimens. Most of the curves seem to show a discontinuity at the centre, position 3. Readings taken close to the centre might show that the curve is really continuous instead of coming to a sharp point.

The curves referred to above represent the sum of the readings in corresponding positions on the two cover plates and thus give (as shown in §2) the mean strain in the cover plates multiplied

FIG. 9.



Distribution of strains in specimens A and C at a load of 30,000 lbs.

by four. Fig. 10 shows, to the same scale as Fig. 9, the readings on the two cover plates for specimen A at 30,000 pounds load. Owing to bending in the plates, and slight irregularities in the placing and action of the rivets, the strain in one cover plate is at each section higher than in the other, the ratio varying somewhat at different sections. This has a slight effect upon the partition of load between the rivets, as was proved in Part I, § 3. The difference in the value of E for the cover plates and the middle plates has a similar effect.

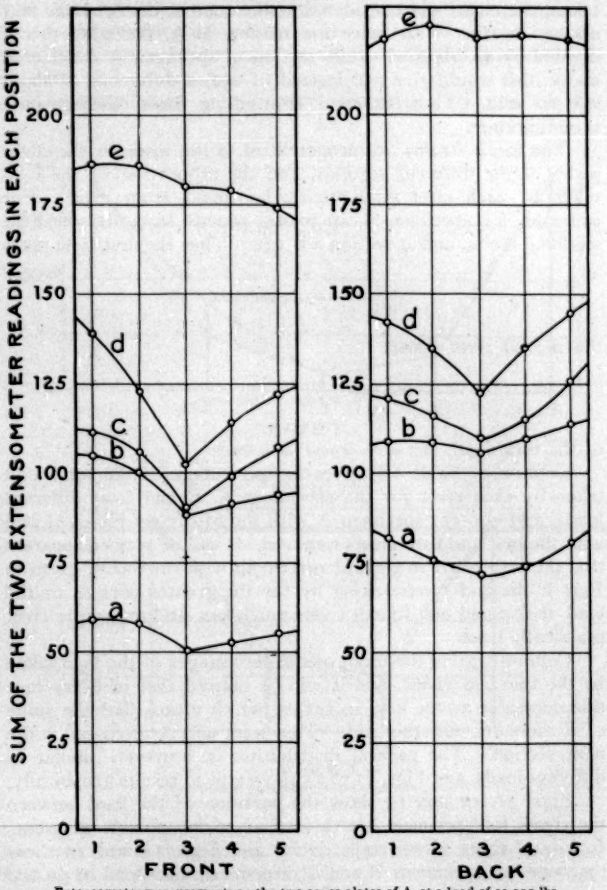
The curves given in Fig. 9 show that the ratio of maximum to minimum stress, or strain, across the section is least between the first and second rivets and increases to a maximum between the fourth and fifth rivets. This fact and the general form of the curves lend strong support to the view that the rivets are acting in shear, because, if the plates were, in the neighborhood of the rivets, held together by friction, it would seem that the pull transferred by each rivet would give a maximum stress at the centre line and a minimum at the outer edges. Also, it is very unlikely that in such a case the strains on the outer faces of the cover plates would give true values for the mean strains in the plates.

The area included between the curve for section e, the ordinates at its ends, and the horizontal axis is a measure, to some scale, of the total load on the specimen. The areas under curves a, b, c and d represent to the same scale the loads at the corresponding sections of the cover plates. Thus the area under a represents the load transferred to the cover plates by the first rivet, the area between a and b the load transferred by the second rivet, etc. It is at once evident that the first and last rivets transfer the major portion of the load and that the middle rivet transfers practically nothing. This is in accordance with the theory given in Part I. In order to obtain the exact loads taken by each rivet, all that is necessary is to obtain the mean heights of the curves. This has been done, and the results are given in Table VI, column 3. It was found that the mean corresponding to the curves could best be determined by using Simpson's rule on each side of the centre. Thus, for example, the mean for specimen A at section d for a load of 30,000 pounds is given by

$$\frac{281+4\times258+2\times226+4\times249+267}{12}=252,$$

the result being taken to the nearest integer.

FIG. 10.



Extensometer measurements on the two cover plates of A, at a load of 30,000 lbs.

For specimens B, D, E and F, since three readings only were taken, the mean was considered as the sum of the readings at 1 and 5, together with twice the reading at 3, the whole being divided by 4. Applied to the section of specimen A considered above, this would give 250 instead of 252, a difference of about 0.8 per cent. Thus the error from using three readings only is unimportant.

The mean strains are proportional to the loads in the cover plates at the different sections, and the proportion of the load taken by each rivet may readily be found from these. For example, for specimen A at 30,000 pounds load, the mean at section e is 404, and at section a is 131. Thus the first rivet takes

$$\frac{131}{404} \times 100 = 32.4 \text{ per cent.},$$

the second rivet takes

$$\frac{204-131}{404} \times 100 = 18 \text{ per cent.},$$

of the total load, etc.

Column 4, Table VI, gives the percentage of the total load taken by each rivet for the specimens A, B and C at different loads, and also for specimen A with the fifth rivet removed and with the first and fifth rivets removed. It will be at once apparent that the results are in general agreement with the theory given in Part I, the end rivets taking by far the greater portion of the load, the second and fourth rivets much less, and the centre rivet practically none.

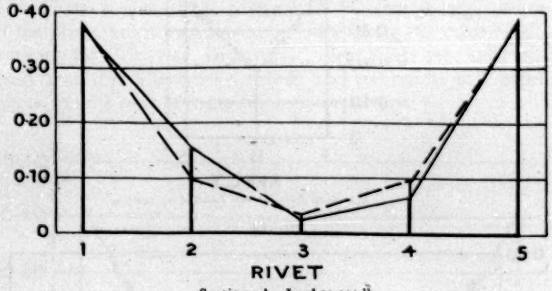
Column 5 gives the mean of the percentages of the load taken by the two end rivets, and it will be noticed that in every case this decreases as the load increases, which means that the value of K increases with the load. This point will be discussed in the next section. The general distribution is, however, similar at different loads, and Figs. 11 to 15 show typical results graphically.

Figs. 11, 12 and 13 show the partition of the load between the rivets for specimen A at a load of 20,000 pounds with five, four, and three rivets respectively, and Figs. 14 and 15 show the same for specimens B and C respectively at a load of 30,000 pounds.

The heavy ordinates represent the proportions of the total load taken by each rivet. The tops are joined by lines in order to display the results more clearly. These lines do not, of course, mean that interpolations may be made for a different number of rivets.

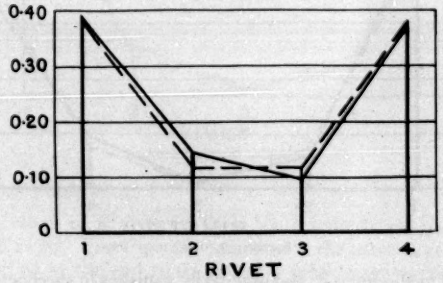
The distribution is not quite symmetrical in any of the

FIG. 11.



Specimen A. Load 20,000 lbs.

FIG. 12.

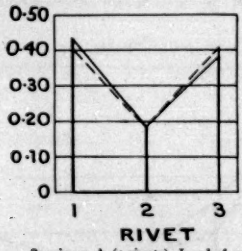


Specimen A (4 rivets). Load 20,000 lbs.

specimens, the last rivet, except for the three highest loads on specimen A and the two highest on the same specimen with the fifth rivet removed, taking a little less of the load than the first. This is, probably, mainly due to the causes discussed in Part I,

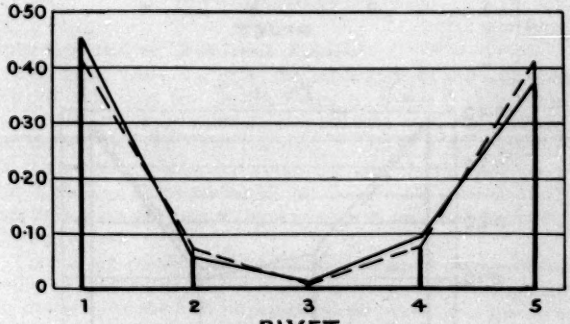
§ 3, *i.e.*, the difference in the values of E for the main plate and the cover plates and the unequal loads taken by the two cover plates; but want of straightness and minor irregularities in the specimens may also play some part. The difference is most

FIG. 13.



Specimen A (3 rivets). Load of 20,000 lbs.

FIG. 14.

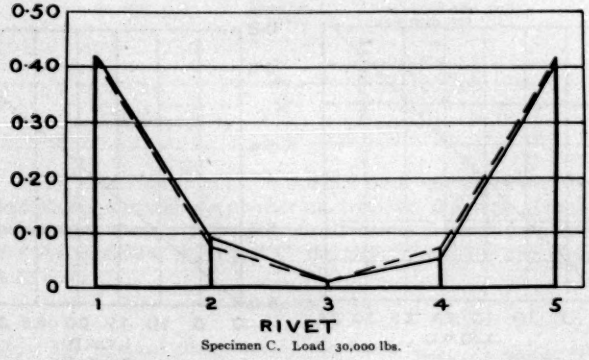


Specimen B. Load 30,000 lbs.

marked in specimen B, and practically vanishes in specimen C. In order to determine the value of K best fitting the experimental results, the mean of the loads taken by the first and last rivets was used, and from this K was determined, as described in Part I, by the use of Fig. 2, Part I. The values thus found are given in the last column of Table VI.

The dotted lines in Figs. 11 to 15 show the theoretical percentages of the load taken by each rivet, obtained by using these values of K. The agreement between the theoretical and experimental results is very striking, especially if it be remembered that the specimens were by no means ideal, but ordinary shop products. The experimental results for the rivets 2, 3 and 4 are somewhat irregular. This may be partly owing to experimental errors, since the results depend upon the comparatively small differences between the means of the readings at consecutive cross-sections of the plates, but it may also arise from taking the value of K as constant for all rivets. In any case, these rivets take such small loads that the differences between the theoretical and experimental results may be regarded as unimportant.

FIG. 15.

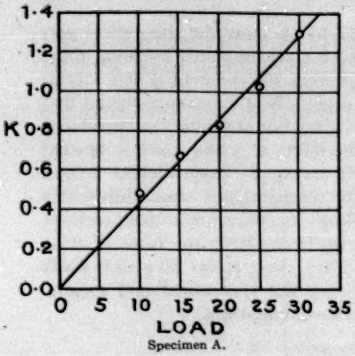


Specimen C. Load 30,000 lbs.

§ 4. *The Value of K.*

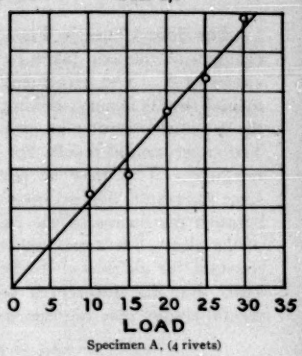
The values of K given in Table VI, column 6, are plotted in Figs. 16 to 20. Since a small variation in the percentage of the load carried by the end rivets causes a fairly large variation of K, the results cannot be considered as accurate to the third place of decimals, and the range of error is roughly indicated by the circles marking the experimental points in the figures. The variation of K with the load is somewhat irregular, but the points lie fairly well on straight lines except for specimen C. Probably

FIG. 16.



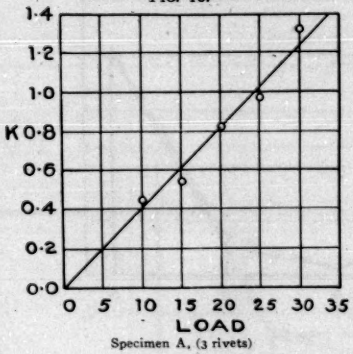
Specimen A.

FIG. 17.



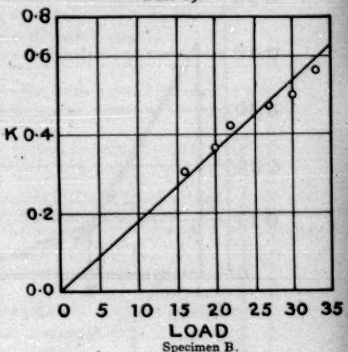
Specimen A, (4 rivets)

FIG. 18.



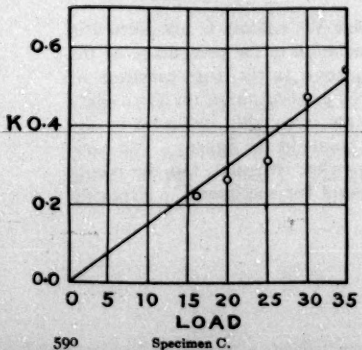
Specimen A, (3 rivets)

FIG. 19.



Specimen B.

FIG. 20.



Specimen C.

FIGS. 16–20. Variation of K with load. Unit of load–1000 lbs.

with more carefully made specimens the irregularities would disappear. In any case the law of variation may be regarded, for practical purposes, as linear. In order to obtain the mean straight lines, the method of least squares was used; *i.e.*, if F_n represent the abscissa (load), K_n the ordinate (K) of any point, the law of variation was taken as

$$K = \frac{\Sigma F_n K_n}{\Sigma F_n^2} \cdot F.$$

The results are given, correct to two figures, in the annexed Table.

TABLE VII.

Specimen	Diameter of rivets	Law of variation of K with load
C	⅞″	$K = 0.15 \times 10^{-4}$ F
B	¾″	$K = 0.18 \times 10^{-4}$ F
A	½″	$K = 0.43 \times 10^{-4}$ F
A (4 rivets)	½″	$K = 0.44 \times 10^{-4}$ F
A (3 rivets)	½″	$K = 0.41 \times 10^{-4}$ F

The results differ but little for the three specimens with ½-inch rivets, although the number of rivets was different. On the other hand, there is a considerable difference between the values for the specimens A, B and C, the ratio of K's for any given load being

$$K_A : K_B : K_C = 2.87 : 1.2 : 1.$$

Now the nominal diameters of the rivets in A, B and C are ½ inch, ¾ inch, and ⅞ inch respectively, and the inverse ratio of their areas is as

$$3.07 : 1.34 : 1$$

Thus the values of K are roughly in inverse proportion to the nominal areas. A number of rivets were removed from these specimens, and it was found that the rivets fitted tightly in the holes, while the holes were somewhat irregular but always greater in diameter than the nominal diameter of the rivet, the mean diameters being 0.55 inch for specimen A, 0.83 inch for specimen

B, and 0.925 inch for specimen C. Thus the inverse ratio of the actual areas is as

$$2.83 : 1.24 : 1,$$

which is very close to the ratio of K's. It therefore appears that the values of K vary as $\frac{F}{A}$, where F is the total load on the specimen and A the area of cross-section of the rivets. This being so, an empirical formula for K is

$$K = \frac{a F}{10^{-4} A},$$

where a is a constant. The values of a given by the experimental results are 0.102, 0.098, and 0.101 for the specimens with ½-inch, ¾-inch, and ⅞-inch rivets respectively. Thus the mean value is 0.100 and

$$K = \frac{F}{100{,}000 A} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (17)$$

where A is the actual area of cross-section of the rivets. This result indicates that in joints similar to the specimens A, B and C the value of K is the same for the same average load per square inch of total cross-section of the rivets. For example, at a load of 10,000 pounds per square inch of total cross-section of the rivets $K = 1.0$. On the other hand, the results for specimen A with the fifth rivet and with the first and fifth rivets removed show very little change in the value of K with the number of rivets. The results of these tests, however, cannot be taken as conclusive, because the rivets had been previously under stress in the complete specimen.

It will be shown in the next section that the value of K given by equation (17) gives results in accordance with the experimental results for the remaining specimens. This says much for the truth of the theory given in Part I, and shows that equation (17) is correct for specimens such as have been tested. It will be necessary, however, to make many more experiments on specimens in both tension and compression having different sizes and arrangements of rivets and different ratios of width of cover plate to rivet pitch before general rules can be given for the determination of K in any type of joint.

Equation (17) is entirely empirical. It will be interesting to examine the results theoretically. K, as explained in Part I, § 2, is a coefficient given by

$$K = \frac{2a_c}{l} \cdot k.$$

where k is that quantity which, when multiplied by the square of the load transferred by a rivet and divided by $2E$, gives the work stored in the rivet or its equivalent. It is not easy to determine in exactly what manner work is stored in the rivets. Possibly the rivets act by giving a frictional hold between the plates. This does not seem probable, since K follows the same laws below and above loads at which slip has been shown to occur by other investigators, and because of other reasons already given. If, however, the rivets do hold by friction, K must be some function depending upon the work stored in the portions of the plate held together without slip by the rivets. If, on the other hand, the rivets are in shear, $\frac{kP^2}{2E}$ is the work stored in them when transmitting a load P. Now the rivets are so short and so rigidly held that the work stored in bending must be very small and the major portion of $\frac{kP^2}{2E}$ must be the work stored in shear. This will depend upon the exact manner in which the load comes upon the rivets. As the load increases the contact between plates and rivets will become more intimate and thus the value of K will increase. This is precisely what was found in the experiments. It is, of course, impossible to say theoretically how the load will be distributed at any particular stage. Assuming, however, that a load has been reached which gives a uniform load distribution over the length of the rivets, the shearing force on the rivet will increase uniformly from zero at the head to a maximum of $\frac{P}{2}$ at the junction of the cover and middle plates and will then decrease uniformly to zero at the centre of the middle plate. The intensity of shear, q, at any point of a cross-section over which the shearing force is S will be given by

$$q = \frac{4S}{3\pi R^2}\left(1 - \frac{y^2}{R^2}\right)$$

where R is the radius of the rivet and y the distance of the point

from an axis through the centre in the plane of the cross-section and perpendicular to the load.[12] The work stored by shear will be given by

$$W_s = \Sigma \frac{a^2}{2G} \cdot dV,$$

where G is the modulus of rigidity of the material and dV an element of the volume. This, omitting the analysis, gives

$$W_s = \frac{5}{54G} \cdot \frac{t}{A} \cdot P^2,$$

where t is the thickness of the middle plate, or twice the thickness of the cover plates.
Thus

$$K = \frac{5}{27} \frac{E}{G} \cdot \frac{t}{A}$$

and

$$K = \frac{2a_c}{l} \cdot \frac{5}{27} \cdot \frac{E}{G} \cdot \frac{t}{A}.$$

If b represents the breadth of the cover plate $a_c = \frac{bt}{2}$, and

$$K = \frac{5}{27} \cdot \frac{E}{G} \cdot \frac{b}{l} \cdot \frac{t^2}{A}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (18)$$

For specimen A, having a middle plate 3 inches wide and ⅝-inch thick and ½-inch rivets of 4 inches pitch, this would give, taking the nominal area of the rivets and $\frac{E}{G} = 2.5$,

$$K = 0.692.$$

This is of the same order as the experimental values; in fact, it is the actual value of K for a load of 16,000 pounds or an average load of 8150 pounds per square inch of rivet. If the work stored in bending were considered, the value would be raised somewhat. The value of K given by equation (18) varies inversely as A, and this would be true no matter how the load was distributed over the length of the rivet. This is in accordance with the experimental results. On the other hand, it varies as t^2 while the experimental results do not show such a variation. This would appear to indicate that at a given load per square inch of rivet

[12] See Morley, "Strength of Materials," p. 132. This assumes that the shear intensity is constant over an elementary slice of the cross-section perpendicular to the load.

the load is distributed in the same way and over the same length of the rivet, no matter what the total length of the rivet may be. This seems probable, at any rate for fairly short rivets. In this case K would vary with the ratio $\frac{b}{l}$. Further theorizing on this point, however, would be futile on the experimental evidence at present available.

To sum up, if the rivets act by clamping the plates together by their initial tension, it is evident from the experimental results of §2 that this action is local, and in this case K will depend on the way in which work is stored in the parts of the plates thus held. If, on the other hand, the rivets are in shear, K will depend principally upon the work stored in shear in the rivets and will vary at different loads, because the manner of distribution of the

TABLE VIII.

Specimen	Section	Sum of extensometer readings on the two faces			Mean	Specimen	Section	Sum of extensometer readings on the two faces			Mean
		1	2	3				1	2	3	
D	a	145	119	143	132	F	a	76	67	77	72
	b	164	138	163	151		b	110	90	109	100
	c	173	144	166	157		c	140	114	142	127
	d	176	144	170	158		d	162	150	166	157
	e	195	170	197	187		e	279	281	281	280
E	a	187	170	192	180	F′	a	144	134	150	140
	b	216	183	212	198		b	175	156	179	166
	c	221	184	223	203		c	197	157	196	177
	d	228	191	226	209		d	215	178	227	199
	e	280	284	281	282		e	...	...	...	280

load on a rivet depends upon the intensity of the load. Theoretical considerations and experimental results both appear to show that the second hypothesis is the correct one, but further experiments are needed. It is, of course, quite possible that the value of K is not the same for all the rivets in a joint and that the values given above are only equivalent ones for the whole joint.

§5. *The Specimens D, E, F and F′.*

The experimental results for the specimens D, E, F and F′ under a load of 16,000 pounds are shown in Tables VIII and IX. The specimen F′ was, as mentioned above, obtained by cutting

down the middle plate of specimen F to a uniform thickness of 4.096 inches. This, of course, changed the partition of the load among the rivets, as may be seen by comparing Figs. 23 and 24.

The specimens D, E, and F′ will be considered first. The cover plates in specimens D and E were ⅜ inch and ¼ inch thick respectively, the middle plate being ¼ inch thick, and all plates 3 inches wide. All the plates of F′ were ¼ inch thick, but the outer plates were 3 inches wide and the inner plate 4.096 inches. Thus the values of C in the specimens D, E and F′ were 4, 3 and 2.468 respectively. Now it will be remembered that the theory given in Part I indicated that, if C were greater than 2, the first rivet would take a greater percentage of the load than the last,

TABLE IX.

Specimen	C	K		Percentage of total load taken by each rivet				
				1	2	3	4	5
D	.....	....	e	70.7	10.1	3.2	0.5	15.5
	4.04	0.29	t	70.5	4.4	0.4	1.5	23.2
E	.....	....	e	63.8	6.4	1.8	2.1	25.9
	3.02	0.29	t	61.5	5.0	0.2	2.9	30.4
F′	.....	....	e	50.0	9.3	3.9	7.9	28.9
	2.468	0.29	t	53.8	5.2	0.6	3.8	36.6
F	.....	....	e	25.7	10.0	9.6	10.7	44.0
	2.468	0.29	t	28.7	8.0	6.6	11.6	45.0

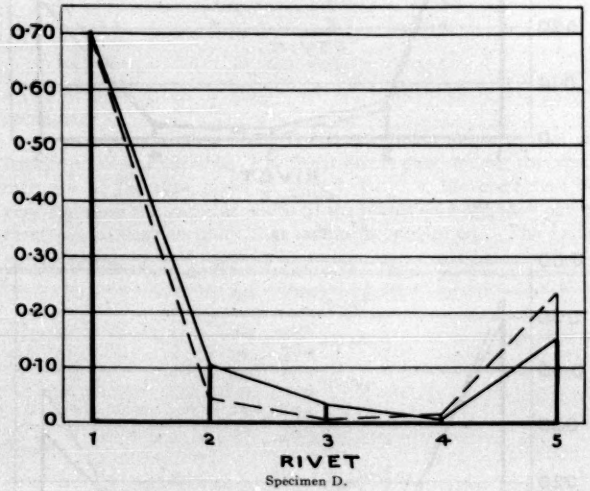
e = Experimental.
t = Theoretical.

the difference increasing with the value of C. This is entirely borne out by the experimental results as shown in Table IX, the percentage taken by the first rivets being respectively 70.7, 63.8, and 50.0 for the three specimens. The loads taken by the other rivets also agree in general with the theoretical results.

It was shown in the last section that K for ¾-inch rivets is equal to 0.18×10^{-4} F. For a load of 16,000 pounds this gives $K = 0.29$. If, as surmised above, a small difference in the length of the rivets does not alter K, the above value ought to give figures which agree with the experimental results. The two cover plates, however, did not in any of the specimens receive exactly the same loads, and allowance must be made for this, as explained in §3, Part I. The actual ratio of the loads taken by the two cover plates was not constant at each section. This could be

allowed for if necessary, but the effect of the correction is, in any case, so small that it will be quite sufficient to take mean values. The mean values of $\frac{1}{s}$ in D, E, and F were 0.595, 0.574, and 0.510 respectively. Thus, allowing for these, the values of C become 4.04 and 3.02 for D and E respectively, remaining practically unchanged for F'. By substituting these values in the equations of Part I and taking $K = 0.29$, the figures in the rows marked t in Table IX were obtained, and the results are shown with the experimental results in Figs. 21 to 23. It will be seen that the agreement is fairly close. It is almost exact for the first rivets,

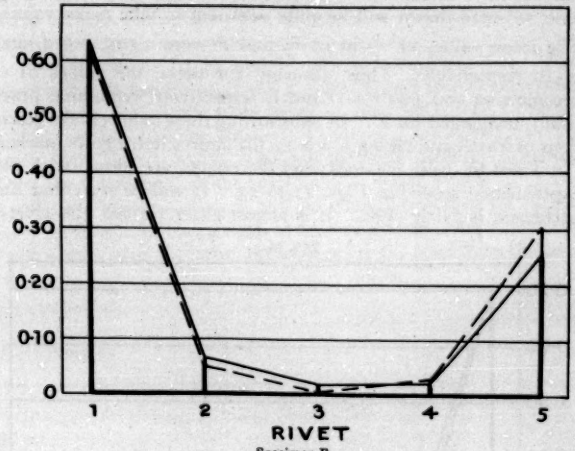
FIG. 21.



Specimen D.

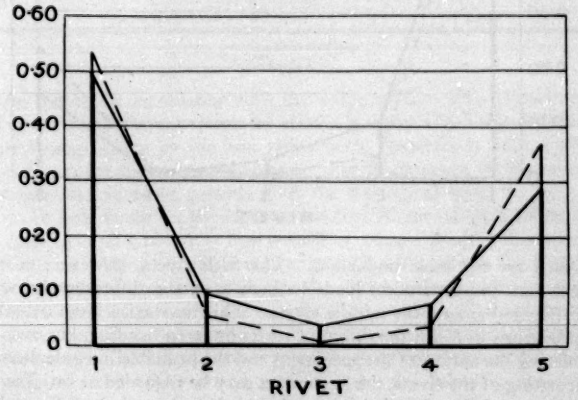
which are the most important. The fifth rivets, however, take less than the theoretical loads in each case, the difference being distributed among the middle rivets. This may arise from actual differences in K for the different rivets or from bending, but, considering the nature of the specimens and the probable irregularities in setting of the rivets, the agreement may be regarded as satisfactory, especially when it is remembered that the value of K used

FIG. 22.



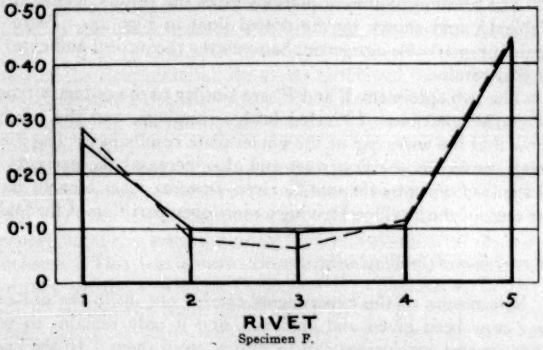
Specimen E.

FIG. 23.



Specimen P'.

FIG. 24.



Specimen F.

was obtained from a single set of experiments on another specimen.

F was the only specimen tested in which the width of the middle plate was variable. The theoretical equations for this specimen are of the type given in Part I, §4. η, the coefficient depending upon the variable width of the plates between each pair of rivets, is so near to unity that it may be neglected. The values of the terms in $2\Sigma\frac{a_c}{a_1}$ are

$$\frac{2a_c}{a_1}=\frac{2\times3}{12.32}=0.487,$$
$$\frac{2a_c}{a_2}=\frac{2\times3}{10.2}=0.588,$$
$$\frac{2a_c}{a_3}=\frac{2\times3}{8.08}=0.743,$$
$$\frac{2a_c}{a_4}=\frac{2\times3}{5.96}=1.007.$$

Thus

$$2\sum_{1}^{4}\frac{a_c}{a}=2.825,\ 2\sum_{2}^{4}\frac{a_c}{a}=2.338,\ 2\sum_{3}^{4}\frac{a_c}{a}=1.750$$

and $2\sum_{4}^{4}\frac{a_c}{a}=1.007$, and the equations for the loads taken by the various rivets, Part I, §4, p. 589, taking $K=0.29$, become

7.405	5.628	4.040	2.297	3.115
5.628	5.918	4.040	2.297	2.628
4.040	4.040	4.330	2.297	2.040
2.297	2.297	2.297	2.587	1.297

The solution of these equations gives the results displayed in Table IX and shown by the dotted lines in Fig. 24. There is again a remarkable agreement between the theoretical and experimental results.

The two specimens F and F′ are similar to the ordinary types of end connections of riveted bridge members, and the results show that the widening of the gusset plate results in far less load being carried by the first rivet and also increases the proportion of the load taken by the middle rivets from 21.1 per cent. to 30.3 per cent. of the total load, giving a more even partition of the load.

§6. *General Conclusions.*

The results of the experiments carried out up to the present have now been given and analyzed, and it only remains to see what general conclusions can be drawn from them. In the first place, all the experiments are in remarkable agreement with the theory advanced in Part I, especially when it is taken into consideration that the specimens were ordinary shop products, and show that it is possible to predict, in general, the way in which the load will be divided among the rivets in any form of joint.

Only the specimen with ½-inch rivets was carried beyond the working load, but the regularity of its action showed that the partition of load obeyed the same laws at all loads up to that causing permanent deformation of the plates or rivets. In every specimen and at all loads the first and fifth rivets took by far the greater part of the total load, the actual proportion decreasing gradually as the load increased. For example, in the specimen with ½-inch rivets, the first and fifth rivets carried 83.5 per cent. of the total load at a load of 10,000 pounds, and this decreased to 70.0 per cent. at a load of 30,000 pounds. The latter load corresponds to an average stress of about 12,650 pounds per square inch of actual rivet section, or 15,280 pounds per square inch of nominal rivet section. This would usually be taken as the shearing stress on all the rivets, but actually the end rivets, if, as there seems little doubt, the rivets were in shear and not holding by friction, were each under an average shear stress of 22,150 pounds per square inch, while the third rivet at the same load took only 3.2 per cent., corresponding to an average shear stress of only 2020 pounds per square inch. Thus in joints having

several rows each containing an equal number of rivets and designed in the usual manner, *i.e.*, allowing the average load per square inch of rivet section to be equal to the working stress in shear of the rivet material, the rivets in the end rows must carry stresses far above the allowable working stress. That this will not be remedied by increasing the number of rows of rivets was shown in Part I, §2.

The above refers to the specimens in which the cover plates were of the correct thickness; *i.e.*, each half as thick as the middle plate. If they are thicker, the first rivet takes an even greater proportion of the load, the proportion increasing with increased thickness. This was shown theoretically in Part I and experimentally on the specimens D, E and F'. Specimens B, D, E and F' all had the same diameter of rivet. In specimen B, in which the cover plates were of correct thickness, the first and fifth rivets took 88.1 per cent. of the load at a load of 16,000 pounds. In specimens D, E and F at the same load they carried 86.2 per cent., 89.7 per cent. and 78.9 per cent. of the total load respectively, but of these the first rivets carried respectively 70.7 per cent., 63.8 per cent. and 50 per cent. of the total load. Specimen F, in which the middle plate was of varying width, illustrated the action in members riveted to a gusset plate, and it was found that the varying width of plate resulted in a rather more even distribution of stress, the first and fifth rivets carrying only 69.7 per cent. of the load, as compared with 78.9 per cent. when the middle plate was cut down to uniform width.

It must be noted that in all the specimens tested the ratio of width of cover plate to pitch of rivets was the same, ¾. Now it was shown in §4 that K probably varies as the width of cover plate divided by the pitch of the rivets; thus, with a smaller pitch or wider plates, K would be increased, and the effect of this would be to make the partition of the load rather more uniform. But, as stated above, a large variation of K only causes a comparatively small alteration in the percentage of the load carried by the end rivets. For example, in the specimen A a change of K from 0.485 to 1.3 only altered the load carried by each of these rivets from 40.7 per cent. to 35 per cent., and the alteration for a given change becomes less and less as the values of K increase. Thus the effect of change of pitch or breadth of cover is not likely to be very great, except possibly in splices containing a number of

rivets in each row. However, further experiments are needed in order that a general law may be found for the value of K. When this is determined it will be possible to predetermine the exact partition of load in any proposed joint, and this will enable the joint to be designed in the most efficient manner. A very good approximation, sufficient for most purposes, may, however, be obtained from the data already given, since the general manner of partition of load is the same for all values of K. It would require too much space to illustrate this further here, but the examples given in Part I show clearly the method of procedure.

The writer has already in hand further experiments on the variation of K in different types of joints and also experiments designed to show the part, if any, played by friction in riveted joints.

§7. *Summary and Conclusion.*

The following is a summary of the principal contents of the present paper:

1. It is shown that a riveted joint may be considered as a statically indeterminate structure, and that a series of equations may be obtained for any joint by means of the Principle of Least Work, giving the loads carried by each of the rivets in terms of a quantity K, which depends upon the manner in which work is stored in, or by the action of, the rivets.

2. This theory is applied to various types of joints, and the modifying effects of non-uniform distribution of stress in the plates, unequal partition of the load between the two cover plates, and a difference in the modulus of elasticity of the middle plate and the cover plates are also considered.

3. It is shown experimentally that extensometer measurements on the outer surfaces of the cover plates of a riveted joint are sufficient for the determination of the mean stresses in the plates, and that the partition of the load among the rivets may be determined from such measurements. It is also shown that, at any rate after the first few loadings, the distribution of strain in the plates of a joint is not altered by repeated loadings.

4. It appears from 3 that if there is any frictional hold between the plates, it acts only over those portions in the immediate neighborhood of the rivets. All the experiments tend to show that friction does not play an important part, but further experiments are necessary on this point.

5. Experiments made on a number of specimens having a single line of rivets and loaded in tension give results in close agreement with the theoretical considerations. They also show that the longitudinal stresses in a portion of the cover plate between two consecutive rivets are a minimum along the line of rivets, rising to a maximum at the edges of the plates.

6. The experiments show that the value of K for a joint having a given ratio of width of cover plate to rivet pitch and a given number of rivets varies approximately directly as the load and inversely as the area of the rivets. An empirical rule is given for its value in joints similar to the experimental specimens, but a more general rule cannot be given until further experiments have been made. A theoretical estimate is made of the value of K for a rivet acting in shear, and the result is shown to be within the range of the experimental values.

7. Both the experimental results and the theoretical deductions show that :—

(*a*) in a double-cover butt joint having a single line of rivets, the two end rivets and the two rivets on each side of the junction of the middle plates take by far the greater part of the load at all loads within that causing permanent deformation of the plates or rivets, the actual proportion decreasing slowly as the load increases;

(*b*) if, in such joints, the total area of cross-section of the cover plates is equal to that of the middle plates, these four rivets take equal loads, but if it is greater the end rivets take greater loads than the others, the difference increasing as the area of the cross-section of the cover plates increases;

(*c*) if two plates of uniform width and equal thickness are connected by a single line of rivets to opposite sides of a gusset plate of uniform width, the first and last rivets take the greater part of the load, but if the gusset plate increases in width from the first to the last rivets, the partition of load is more uniform.

The results already obtained allow the general manner of partition of load in any riveted joint, in which there is no eccentricity of connection, to be estimated, and it is hoped that, when further experiments have given general laws for the value of K, it will be possible to predetermine the exact load that will be carried by each rivet. The practical value of this is obvious.

In conclusion, the writer wishes to thank Prof. H. M. Mackay and Prof. E. Brown for their personal interest and advice, and Mr. S. D. Macnab for his valuable assistance in the experimental part of the work.

McGill University,
August, 1916

Maintenance Cost of Four-cylinder Engines. F. E. WATTS. (*Society of Automobile Engineers Bulletin*, vol. x, No. 5, August, 1916.)—It is an axiom in all mechanical work that the less complicated a machine is, the lower must be its maintenance cost. This is apparent enough in a comparison of the four-cylinder engine with other types. It is more accessible and has fewer parts in proportion to its fewer cylinders. Repairs are consequently not only less frequent but are less costly to make. The fuel efficiency is also fundamentally better. In any type of fuel distribution used in motor cars to-day, the division of the gaseous stream decreases efficiency. The simple cylinder is the most economical type; and the fuel efficiency decreases as the number of cylinders increases. Certified tests made on carefully tuned and adjusted cars are often misleading, and the result in the hands of the user is the only real criterion.

Designers of multi-cylinder engines are now at work on much smaller units utilizing very high compression. This method was tried out about ten years ago and finally abandoned, but with the increased technical skill available at the present time, and the smaller cylinder sizes, it is possible that something can be accomplished. It is, however, doubtful just how much can be done with our present motor car fuel, for the gasoline now available tends to partake more and more of the nature of kerosene, and does not lend itself exceptionally well to high compression. Recent tests have shown that it takes a considerable increase in compression to insure a slight increase in the mean effective pressure, particularly at the lower speeds. It was also found that the engine with high compression does not operate as smoothly as the one with the lower compression, while engines having very high compression may show better fuel economy during tests in the hands of careful users, it does not promise to prove satisfactory for general use.

QUANTITATIVE MEASUREMENTS AT WASHINGTON ON THE SIGNALS FROM THE GERMAN RADIO STATIONS AT NAUEN AND EILVESE.*

BY

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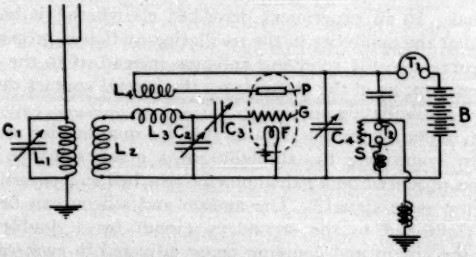
SINCE September, 1914, the strength of the signals received from Nauen and Eilvese has been measured almost daily at the U. S. Naval Radio Laboratory at the Bureau of Standards.

The antenna at the laboratory has an effective height of thirty metres. It is a flat top, 450 feet long, having two wires about two metres apart, 59 metres high at one end and 17 metres at the other. Its resistance at the longer wave-lengths is considerably lower than the one formerly used.

The circuits used for reception are shown in Fig. 1. The

FIG. 1.



detector is the de Forest audion, with an extra coupling, L_3L_4, for producing local oscillations. The current-flow in the audion is naturally unstable, as in the Poulsen arc. This instability tends to produce oscillations, the period of which is determined by the circuit $L_2L_3C_2$. These oscillations are strengthened in the circuit PC_4F, and this increased energy is returned to the circuit $L_2L_3C_2$ through the coupling L_3L_4, thus sustaining the original oscillations, which otherwise tend to break.

* Communicated by the Author.

In the reception of signals from stations sending out continuous oscillations from a high-frequency machine or arc, the receiving audion circuit is so adjusted that the local audion oscillations are slightly out of tune with the incoming oscillations. The combination of the two sets of oscillations produces beats of a frequency which can be heard in the receiving telephones. The pitch of the beats can be adjusted at will by regulating the amount of detuning.

The strength of signal is measured by the shunted telephone method, in which a non-inductive shunt s is placed across the telephones T_2 of the audion and reduced until the signals just remain audible. If I is the value of the current-pulses in the unshunted telephones, and I_0 the current which is just audible,

$$\frac{I}{I_0} = \frac{t+s}{s}$$

where t is the effective telephone resistance for the given beat frequency and s the resistance of the shunt which just permits signals to be heard. The ratio $\frac{I}{I_0}$ is called the audibility of the signal. In an experiment described elsewhere [1] it has been shown that the audibility in the oscillating audion is proportional to the current in the receiving antenna, instead of to the square of the current, as in the electrolytic, the crystal contact detector, and the non-oscillating audion.

The oscillating audion is calibrated for quantitative measurements by comparing the audibility of a given signal with the deflection produced on a galvanometer attached to a silicon detector by the same signal.[2] The audion and silicon can be alternately connected to the secondary circuit by a double-throw switch, the circuit and coupling being adjusted in each case for maximum effect. Just before each experiment the silicon detector is calibrated by comparison with a thermo-element in the artificial antenna. It has been found that when the circuits are properly adjusted for each individual bulb practically all de Forest bulbs which are not evidently imperfect give about the

[1] *Journal Washn. Acad.*, 6, p. 81, 1916.

[2] On account of atmospheric disturbances this calibration is made on an artificial antenna having the same constants as the real antenna, the signals being produced by a second oscillating audion in the laboratory.

same sensitiveness. The sensitiveness, however, varies very greatly for slight changes of adjustment, so for measurement purposes it is of the greatest importance that the adjustment be made in exactly the same way. One of the most reliable methods is to tune the antenna and closed circuits with loose coupling and then, leaving the antenna unchanged, bring up the secondary and retune it at the best coupling. Using Baldwin telephones of 2000 ohms direct-current resistance and a sensitiveness of 5×10^{-10} ampères at 1000 cycles for normal ears, the least audible signal represents about 1.5×10^{-15} watts in the receiving system. The audibility increases as the square-root of the received energy. The proper adjustment of the main coupling appears to make the sensibility within wide limits practically independent of the resistance of the secondary and of the wave-length.

The results of the observations on Nauen and Eilvese from January 1, 1916, to July 1, 1916, are shown in the curves of Fig. 2. The earlier observations are not given, since, on account of changes in the methods of observation, they are not considered properly comparable with the later ones.

The most striking thing about these curves is their remarkable variability, the received current varying from 1×10^{-7} ampères to 80×10^{-7} ampères; that is, from about 25 audibility to more than 2000 audibility, with the method of adjustment used. The cause of these variations is not entirely clear. In some earlier experiments [8] it was thought that probably the variations were, to a considerable extent at least, due to observational errors, but it seems now fairly certain that under normal conditions of atmospheric disturbances the limits of observational error are 20 to 30 per cent., while in extreme cases the limit may be two to one, which will not go far toward explaining these large changes in observed current.

It is now believed that the variation in intensity is due, for the most part, to an irregular reflection or possibly refraction in the upper atmosphere, which brings to the receiving station energy in addition to that which travels along close to the surface of the earth. It is also probable that there is a certain amount of absorption due to the ionization of the atmosphere. Marconi has stated that between Glace Bay and Clifden, Ireland, the signals are weaker during the presence of marked low barom-

[8] *Bulletin Bureau of Standards*, 7, p. 315, 1911.

eter areas between the stations, and the present work seems to support his statements to some extent. There seems to be some reason to believe that regions where atmospheric disturbances are pronounced may be also regions of absorption.

Professor Sommerfeld and his students have developed a theory of the transmission of waves over a good conducting surface like salt water, which may be approximately expressed by the following formula:[4]

$$(1) \qquad I_R = 377 \frac{h_1 h_2 I_s}{\lambda d R} \cdot e^{-\frac{0.0019 \cdot d}{\sqrt{\lambda}}}$$

where I_r is the current in the receiving antenna, I_s the current in the sending antenna, h_1 and h_2 the respective heights of the centres of capacity of the sending and receiving antennas, λ the wave-length, d the distance, and R the resistance of the receiving system. The currents are expressed in ampères and the lengths in kilometres. The first term of this expression is derived from the Hertzian theory. The exponential term represents the falling off in intensity due to the failure of the waves to follow the curvature of the earth. Equation (1) takes no account of the possible return of some of this lost energy from the upper atmosphere, and it seems probable that this equation is fairly correct for the signals on the worst days. The formula for received current used by the Navy Department makes use of the same Hertzian term as Equation (1), but employs an exponential term which was derived empirically from the observations made during the long-distance tests between Brant Rock and the cruisers *Salem* and *Birmingham* in 1909 and 1910.[5]

(2) Navy formula $$I_R = 377 \frac{h_1 h_2 I_s}{\lambda d R} \cdot e^{-\frac{0.0015 \cdot d}{\sqrt{\lambda}}}$$

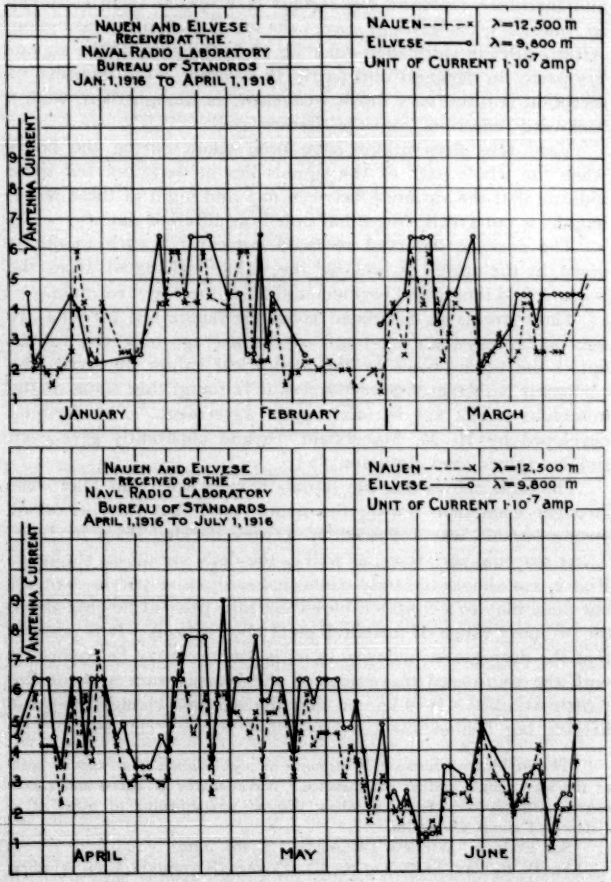
This formula was intended to represent the average value of received current over salt water in the day time, and appears, so far as can be certainly determined, to give results which are correct within the limits of observational error, at least up to 3600 nautical miles.

The observations shown in the curves of Fig. 2 were taken

[4] A. Sommerfeld, *Ann. der Phys.*, **28**, p. 665, 1900; also H. March, *Ibid.*, **37**, p. 29, 1912; W. von Rybczynski, *Ibid.*, **41**, p. 191, 193; and J. Zenneck, "Lehrbuch der drahtlosen Telegraphie," p. 294.

[5] *Bulletin Bureau of Standards*, **7**, p. 35, 1911, Reprint 159, and *Bulletin Bureau of Standards*, **11**, p. 69, 1914, Reprint 226.

FIG. 2.


NAUEN AND EILVESE
RECEIVED AT THE
NAVAL RADIO LABORATORY
BUREAU OF STANDRDS
JAN. 1, 1916 TO APRIL 1, 1916
NAUEN ---x λ=12,500 m
EILVESE ——o λ=9,800 m
UNIT OF CURRENT 1·10⁻⁷ amp
√ANTENNA CURRENT
JANUARY
FEBRUARY
MARCH
NAUEN AND EILVESE
RECEIVED OF THE
NAVL RADIO LABORATORY
BUREAU OF STANDARDS
APRIL 1, 1916 TO JULY 1, 1916
NAUEN ---x λ=12,500 m
EILVESE ——o λ=9,800 m
UNIT OF CURRENT 1·10⁻⁷ amp
√ANTENNA CURRENT
APRIL
MAY
JUNE

in general between the hours of 9 and 11 A.M., Washington time, and therefore represent the signals transmitted from Germany to America when daylight covers the whole path of transmission. After sunset in Germany,—that is, when the path of the signals lies partly in darkness and partly in daylight,—the intensity of reception is often very much weakened, as though there were a backward reflection from the shadow wall.

Very few observations have been taken during the hours when the whole path of the signals lies in darkness, but these indicate that the variation between day and night at these wave-lengths is not greater than that noted on different days.[6]

The average observed received current for each month is given in the table, as well as the values calculated from the Sommerfeld theoretical formula and from the Navy formula.

The currents as calculated from the theoretical formula are seen to be less than one-tenth of the average observed values, but if the square-roots of the theoretical values of current be compared with the observed curves it is found that some of the minimum points are in fairly good agreement. The formula developed by H. M. Macdonald[7] would apparently give even smaller values than Equation (1).

The table shows that the signals during April and May were stronger even than during the winter, and that this period of strong signals was followed by a very decided drop in June. These seasonal variations, as well as the daily variations shown in Fig. 2, are almost certainly connected with atmospheric changes, but these may to a considerable extent take place at heights above the ordinary range of meteorological observations. It is possible that the decrease in intensity in early summer may be connected with the southward movement of the low-pressure area of the North Atlantic. It is known that the seasonal changes at great heights lag behind the corresponding surface changes.[8] The

[6] The well-known increase in strength of night signals is a function both of the wave-length and of the distance. Wave-lengths of 10,000 and 12,000 metres will probably begin to show distinct strengthening at night at a distance of about 5000 miles.

[7] See Eccles's "Wireless Telegraphy," p. 161, 1916.

[8] See W. R. Blair, *Bulletin of the Mt. Weather Observatory*, U. S. Weather Bureau, vol. 6, part 4, p. 179, 1914.

study of these relationships, while it is of great interest, must wait until more observations are available.

MONTHLY AVERAGES OF OBSERVED RECEIVED CURRENT.

1916	From Nauen, ampères	From Eilvese, ampères
January	$13.2.10^{-7}$	$19.5.10^{-7}$
February	10.4	18.2
March	12.6	20.8
April	22.4	29.0
May	21.6	33.8
June	6.2	8.5
General Average	14.4	21.6

CALCULATED.

From equation (1)	0.95	1.91
From equation (2)	14.3	22.2

DATA FOR CALCULATION.

	I_r Ampères	h_1 Metres	h_2 Metres	d Kilometres	λ Metres	R Ohms	$e = \frac{0.0019d}{\sqrt{\lambda}}$	$e = \frac{0.0015d}{\sqrt{\lambda}}$
Nauen	150*	150	30	6650	12500	124	0.0039	0.058
Eilvese	140*	150	30	6100	9800	93	0.0045	0.052

* As given in 1914.

U. S. Naval Radiotelegraphic Laboratory,
August, 1916.

Heat Losses Through Galvanized Roofing. ANON. (*The Iron Age*, vol. 98, No. 8, August 24, 1916.)—Loss of heat through a roof of galvanized steel sheets is less than through roofing consisting of cement and fibrous material compressed to sheets about ¼ inch in thickness, according to experiments made last year by the National Physical Laboratory at Teddington, England. A model hut was built up of walls of cork slabs, 2 inches thick, and a roof of the new material or of galvanized sheet iron. The air inside was heated by electrical resistance, which was not allowed to get sufficiently hot to set up appreciable radiation, and the air was agitated by a fan. The air temperature inside differed by 30° F. contrary to expectation, the heat losses through the cement, measured per square foot, were 20 per cent. greater than through metal, the reason being that the emissivity loss from the surface and not from the thermal conductivity of the material, is the decisive factor. When the iron was painted black the heat transmission increased, becoming equal to that of the cement; when the cement slabs were painted with aluminum they behaved like the iron.

Allowance, Tolerance and Limit. D. T. HAMILTON. (*Machinery*, vol. 25, No. 2, October, 1916.)—In manufacturing operations the terms "allowance," "tolerance," and "limit" are frequently used interchangeably, although as a matter of fact they represent different quantities. *Allowance* is the difference between the diameter of the shaft and the hole it enters to allow for various classes of fits, such as drive, push, running, force, etc. *Tolerance* is the difference between the maximum and minimum diameters of either the shaft or the hole necessary to tolerate unavoidable imperfections in workmanship. *Limit* is the amount set above or below the nominal diameters of the shaft or the hole in order to limit the amount of error that can be tolerated.

For example, suppose it is necessary to make a shaft with a good running fit in a bushing. The difference between the nominal diameters of the shaft and hole is 0.002 inch. This is known as the allowance. The limits on the diameters of the hole and shaft, respectively, are −0.00025 inch, +0.0005 inch, and ±0.00075 inch, so that the tolerance on the hole is 0.00075 inch, and on the shaft, 0.0015 inch. For a nominal hole of 1 inch and shaft 0.998 inch, this gives a maximum hole of 1.0005 inch and a minimum shaft of 0.99725 inch, a difference of 0.00325 inch; and a minimum hole of 0.99975 inch and a maximum shaft of 0.99875 inch, a difference of 0.001 inch. From this it will be seen that the allowance between the shaft and hole is what is desired but seldom obtained.

The Magnetic Shielding of Large Spaces. E. WILSON and J. W. NICHOLSON. (*Proceedings of The Royal Society*, Series A; vol. 92, No. A-644, September 1, 1916.) The magnetic shielding of a large space is a problem wholly different in practice from that of a small space, and in view of important applications, the efficiency to which such shielding can be raised is a matter of importance. Considerations of mobility of the apparatus and weight of iron required necessitate the solution of the problem of maximum shielding for a given weight of iron, and more than two shells, together with an examination of the utility of lamination.

By the use of 1273 kilograms of high permeability dynamo magnet steel, a field of order so low as $3\ (10)^{-8}$ has been obtained in a space of 30 centimeters radius, an accurate method of measurement designed for fields of lower order being employed. The leakage through air spaces in a magnetic shield has been studied and found to be more important than is usually supposed. Although polarization of the shells still exists to a small extent, it is not of sufficient magnitude to effect any of the conclusions which have been reached. The leakage field can in fact be completely isolated from other fields. It is now possible to examine the behavior of iron under practically no magnetic force.

THE VITAL RELATION OF TRAIN CONTROL TO THE VALUE OF STEAM AND ELECTRIC RAILWAY PROPERTIES.*

BY

WALTER V. TURNER,

ASSISTANT MANAGER, WESTINGHOUSE AIR BRAKE COMPANY.
Member of the Institute.

(*Continued from page 462*)

Piston travel, where the type of rigging permits it to vary, is a function of the *time* or duration of brake application, as well as of the cylinder pressure. For a condition of four inches false piston travel, as shown in Fig. 11, dotted curve *A*, in the upper figure, represents more nearly what the variation in travel with cylinder pressure would be for an actual brake application, for the piston travel will not lengthen out immediately. It takes a certain period of time for the jolting of the cars and trucks to assist the brake shoes to pull down on the wheel treads, as illustrated in Fig. 10, and thereby lengthen the piston travel. This is significant, because the shocks occur in the early stages of a brake application. Curve *B* in the lower figure shows what the condition portrayed by curve *A* means in the way of high cylinder pressures for light brake pipe reductions. At the point (6 pounds brake pipe reduction) where the brake with the ideal condition of no false piston travel whatever is just starting to become effective the single-shoe brake rigging with four inches false piston travel has about 21 pounds cylinder pressure, as shown by curve *B*. Is there any wonder that shocks occur in the long passenger trains of to-day? It is necessary to make at least a six- or seven-pound brake pipe reduction in order to insure that all triple valves apply and that sufficient differential may be set up to release them when desired. In the attempt to put the brakes on lightly and avoid shocks, insufficient reductions are made, with the inevitable result of stuck brakes.

All these things may be summed up in the following:

In modern heavy passenger train service, the single-shoe type

* Presented at a meeting of the Mechanical and Engineering Section held Thursday, April 27, 1916.

of foundation brake gear with inherent false piston travel is responsible for:

1. Rough handling of trains in:
   - (*a*) Starting—violent "taking of slack" necessary to get train under way.
   - (*b*) Slowing down.
   - (*c*) Stopping.

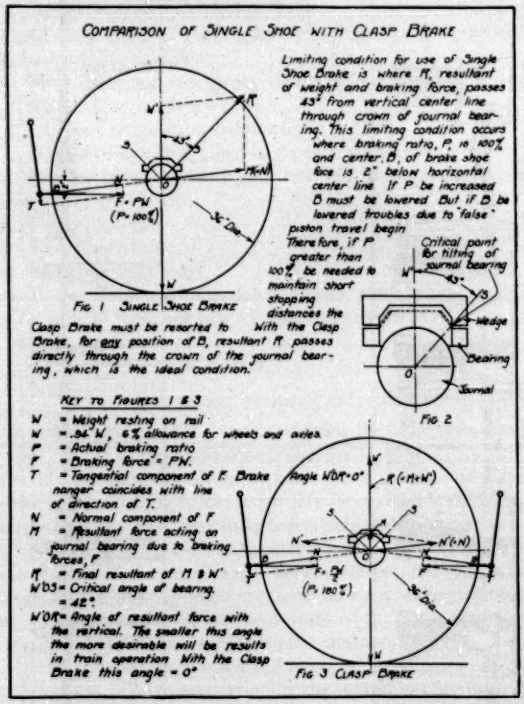
2. Inability to "make the time" because of:
   - (*a*) Hard pulling train—due to dragging brake shoes and stuck brakes.
   - (*b*) Long-drawn-out stops—"dribbling on" brakes in attempt to avoid shocks.
   - (*c*) Delays due to hot journals, stuck brakes, flat wheels, and damage arising from shocks.

3. Unwarranted expense in:
   - (*a*) Excessive fuel and water consumption.
   - (*b*) Reduced capacity of engine.
   - (*c*) Slid flat wheels due to shocks, stuck brakes, and shifting of weight from one pair of wheels to another.
   - (*d*) Damage arising from shocks, even to the extent of break-in-twos.
   - (*e*) Hot journals.
   - (*f*) Burned brake shoes and brake heads.

Obviously, the way to cure these troubles is not to dally with the effects, but to strike back to the underlying causes by applying a suitably designed foundation brake gear of the two-shoe-per-wheel or "clasp" type. Fig. 12 illustrates the part the single-shoe brake plays in giving journal trouble, and the remedy the clasp brake affords. It proves that the clasp brake should be employed whenever it is necessary to exceed a braking ratio of 100 per cent. for either emergency or service applications. And if the point is taken as here established for the failure of the single-shoe brake to be "equal to the job," the need for the clasp brake on account of the overloaded brake shoe will have been cared for long before it arises. This applies to passenger service only, for in freight service, where a comparatively low braking ratio is used,

the absolute value of the brake shoe pressures may, nevertheless, be such as to warrant the use of two shoes per wheel in order to divide the load and reduce brake shoe costs.

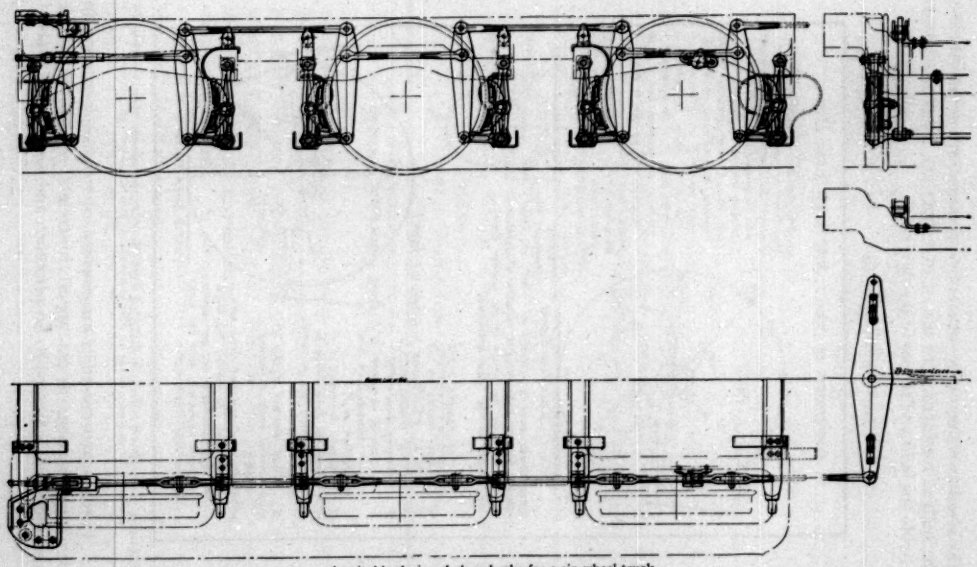
FIG. 12.


COMPARISON OF SINGLE SHOE WITH CLASP BRAKE
Limiting condition for use of Single Shoe Brake is where R, resultant of weight and braking force, passes 43° from vertical center line through crown of journal bearing. This limiting condition occurs where braking ratio, P, is 100% and center, B, of brake shoe face is 2" below horizontal center line. If P be increased B must be lowered But if B be lowered troubles due to "false" piston travel begin
Therefore, if P greater than 100% be needed to maintain short stopping distances the Clasp Brake must be resorted to With the Clasp Brake, for any position of B, resultant R passes directly through the crown of the journal bearing, which is the ideal condition.
Critical point for hitting of journal bearing
Wedge
Bearing
Journal
FIG 2
FIG 1 SINGLE SHOE BRAKE
KEY TO FIGURES 1 & 3
W = Weight resting on rail
W' = .94 W, 6% allowance for wheels and axles.
P = Actual braking ratio
F = Braking force = PW'.
T = Tangential component of F. Brake hanger coincides with line of direction of T.
N = Normal component of F
M = Resultant force acting on journal bearing due to braking forces, F
R = Final resultant of M & W'
W'OB = Critical angle of bearing. = 42°.
W'OR = Angle of resultant force with the vertical. The smaller this angle the more desirable will be results in train operation With the Clasp Brake this angle = 0°
Angle W'OR = 0°
FIG 3 CLASP BRAKE

The clasp brake should always be installed when, in order to maintain short stopping distances, it is necessary to use an effective braking ratio of 100 per cent. or more.

For two reasons the foregoing list of troubles, occurring in passenger service, do not appear in freight service: First, the braking ratio is limited to 80 per cent.; and, second, the type of

FIG. 13.



A suitably designed clasp brake for a six-wheel truck.
Proper truck construction must contribute with suitable clasp brake design in the elimination of false piston travel.

truck generally used does not permit relative movement between the truck frame and wheel.

In summing up, it may be said that a well-designed clasp-brake rigging, such as that shown in Fig. 13, eliminates the single-shoe brake evils above scheduled as no other device can possibly do. A more direct comparison may be drawn up between the single shoe and clasp types of brake gear by saying that with the clasp brake it is possible to have:

1. Shorter stops in emergency, due to reduced brake shoe duty.
2. Reduced brake shoe wear.
3. Reduced brake shoe maintenance.
4. No brake shoe dragging—reduced train resistances.
5. Longer trains handled with less loss of time, using same motive power equipment.
6. Fewer delays.
7. Smoother stops.
8. More accurate stops.
9. Fewer slid flat wheels, due to shocks, stuck brakes, and the transfer of load from one pair of wheels to another.
10. Fewer stuck brakes.
11. Fewer hot journal bearings.

BRAKE CYLINDER PRESSURE REGULATOR.

In order to have some measure of relief from the evils attendant upon the use of the single-shoe brake, pending the accumulation of the funds required in order to meet the sometimes very great expense of remodelling trucks to provide for a suitable clasp brake rigging, a device has been designed which makes the auxiliary reservoir volume a function of the piston travel. The brake application starts with an auxiliary of very much reduced size, which keeps down the brake cylinder pressures for light brake pipe reductions. The continued outward movement of the brake piston cuts in, at predetermined points, additional auxiliary volumes, finally giving full service pressure when the proper brake pipe reduction has been made. By so proportioning the parts it has been actually possible to get a characteristic curve of operation like that of curve *C* in Fig. 11. This provides, as is readily seen, even greater flexibility of brake operation than the clasp brake, but, of course, all the troubles and losses due to dragging brake shoes remain. This new device also arranges for the use of the first small auxil-

iary reservoir for all reapplications of the brake after a partial release, thereby cutting down the build-up in pressure on that retained in the cylinder and making smooth handling of long trains at low speeds much more certain.

GRADE SERVICE.

The problem afforded the operating departments of our railroads in the control of freight trains over mountain grades has always been far more difficult of satisfactory solution than the one arising in controlling passenger trains over similar districts. This is obviously because of the comparatively short length of passenger trains and their higher and *constant* braking ratio. The manufacture of the ton-mile will, therefore, occupy our attention in this phase of train control, and the empty and load freight brake will serve well as an introduction.

In grade service the advantages of the empty and load brake are at first glance more apparent than those had for level road service in the elimination of shocks. In descending a grade a certain percentage of the car weight is acting to accelerate it. This portion is equivalent to the percentage grade; that is, on a 2 per cent. grade 2 per cent. of the weight of a car on this grade is tending to accelerate it. Thus, if the car weighs 100,000 pounds, 2000 pounds is acting to accelerate it down the grade, and if this acceleration is not to take place, an equal opposing force must be brought into play to counteract the accelerating force.

The internal friction or resistance of the car provides a part of this opposing force. If this resistance be taken as 5 pounds per ton, a grade of 5/2000, or 0.25 per cent., is required for the accelerating force to just equal the internal resistance. The 2 per cent. grade will cause a net accelerating force of $2.0-0.25=1.75$ per cent. of the car weight. It is as though the car was without any internal resistance whatever and stood upon a 1.75 per cent. grade. In figuring train control on grades it is necessary, therefore, to consider the internal resistance of cars, but this is neglected in figuring stops, for the reason that many tests have demonstrated that in bringing a car to a stop the internal resistance is just about equal to the force required to overcome the rotative energy of the wheels. In the subsequent formulæ given for stop distances you will notice that no allowance is made for either the rotative energy of the wheels or the internal resistance. In controlling a car on a

grade the rotative energy of the wheels does not change, practically speaking, and therefore the internal resistance operates in the direction of overcoming the accelerative force due to grade.

The retarding force set up by the brakes on a car is, in terms of car weight, equal to the braking ratio multiplied by the brake rigging efficiency and by the coefficient of brake shoe friction. Thus if a loaded car equipped with a single capacity brake has a braking ratio of 15 per cent., a rigging efficiency of 85 per cent., and a brake shoe friction of 15 per cent., the retarding force will be $0.15 \times 0.85 \times 0.15 = 0.01915$, or 1.9 per cent. of the car weight; that is, this retarding force will just equal a net accelerating force of 1.9 per cent., or one due to a grade of $1.9 + 0.25 = 2.15$ per cent., making the above allowance for internal resistance. But this would provide no reserve, no margin for safety or flexibility, because it would necessitate a constant cylinder pressure of 50 pounds (the basis for the braking ratio mentioned) all the way down the grade. Leakage, recharging, and the ability to come to a stop when desired would be quite uncared for. Considering time for recharging, length of train, reserve power for making a stop, leakage, etc., the cylinder pressure will average only 30 pounds for the entire descent of the grade. This means that the retarding force will be only 30/50 of 1.915, or 1.15 per cent., and the maximum grade corresponding, $1.15 + 0.25$, or 1.40 per cent. Making the same allowances for the empty and load brake, with its load braking ratio of 40 per cent., we find the maximum grade to be

$$G = 0.40 \times 0.85 \times 0.15 \times 30/50 + 0.25 = 3.31 \text{ per cent.}$$

You will notice that these limiting grades stand in approximately the same relation as do the braking ratios for the two types of brake. In each case approximately the same factor of safety holds good; that is to say, the empty and load brake is as safe on the 3.31 per cent. grade, as the single capacity brake is on the 1.40 per cent. grade, the only difference being an increase in air consumption of 17 per cent. for the empty and load brake.

The empty and load brake will control a train of loaded cars on a certain grade with only *one-half the air consumption* required by the single capacity brake with the same train on the same grade. This, in itself, shows how marvellously more flexible, more safe, and more economical the empty and load brake is.

The superior air consumption performance and reserve braking

force of the empty and load brake may be utilized in several ways. The speed of trains down grades may be increased, the speed restriction or limitation now passing from the brake to the question of wheel temperatures. The length of trains or the loading of cars may be increased, or all these increases made, and still leave the reserve in air supply and braking force comfortably great. And keep in mind the increased traffic capacity, reduced unit costs, and correspondingly increased net revenue that this means.

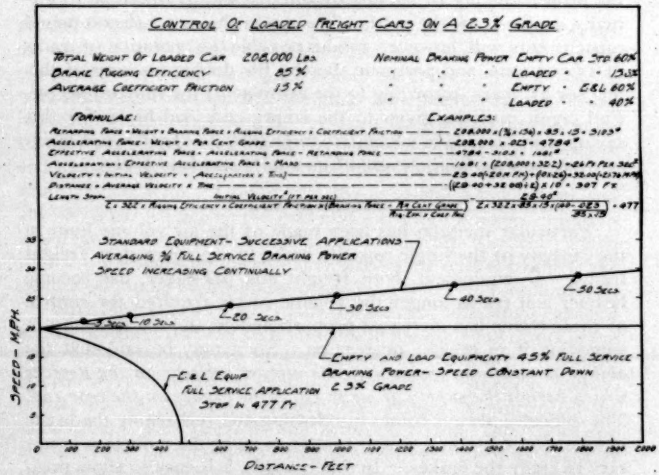
It will be appreciated that the reduction of 50 per cent. in air consumption in providing an equal braking force is of vast importance, that it is of *vital importance*, from a general operating standpoint, when it is remembered that brakes can be operated only by creating differentials on the one hand, and by replenishing the air brake system on the other. And it should be self-evident that to do either increases in difficulty with the increase in quantity of air required. We may say, roughly speaking, that this difficulty is in geometric proportion rather than arithmetic; that is, every time the quantity of air required is doubled, the difficulty will increase four times. Thus it will be seen that the larger the brake cylinders and, consequently, the auxiliary reservoirs that must be employed, the more difficult it becomes to release the brakes and to recharge the brake system. To those who have some idea of what a large auxiliary can absorb from the brake pipe and the effect of the resistance of a long brake pipe to the flow of large quantities of air, what is said above will only add emphasis to their present knowledge. But to those not familiar with or experienced in this problem it may be a revelation to say that with the present 10-inch freight brake equipment we have about reached the limit of our ability to efficiently operate brakes through the medium of compressed air, modern train lengths considered. With this statement properly digested, the importance of employing a brake which for a certain retarding force requires less than half the usual quantity of air will be appreciated.

In the matter of coming to a stop on a grade, the stop distance will be, other things being the same, inversely proportional to the braking ratio; that is, the loaded train equipped with the empty and load brake will stop in 15/40, or 37.5 per cent. of the distance required by the train with the single capacity brake.

Fig. 14 sums up in a graphic way the advantage the empty and load brake offers in grade work. The train equipped with the

single capacity brake, averaging 75 per cent. of the full service braking power (average cylinder pressure of 37.5 pounds—an assumption really too high), is running away on the 2.3 per cent. grade. In less than a minute's time its speed has increased from 20 to 30 miles per hour. The train equipped with the empty and load brake holds to the constant speed of 20 miles per hour with but 45 per cent of the full service braking power (average cylinder

FIG. 14.


CONTROL OF LOADED FREIGHT CARS ON A 2.3% GRADE
TOTAL WEIGHT OF LOADED CAR 208,000 LBS.
BRAKE RIGGING EFFICIENCY 85%
AVERAGE COEFFICIENT FRICTION 13%
NOMINAL BRAKING POWER EMPTY CAR STD 60%
LOADED 13.3%
EMPTY E&L 60%
LOADED 40%
FORMULAE:
EXAMPLES:
STANDARD EQUIPMENT- SUCCESSIVE APPLICATIONS AVERAGING ¾ FULL SERVICE BRAKING POWER SPEED INCREASING CONTINUALLY
10 SECS.
20 SECS.
30 SECS.
40 SECS.
50 SECS.
EMPTY AND LOAD EQUIPMENT 45% FULL SERVICE BRAKING POWER- SPEED CONSTANT DOWN 2.3% GRADE
E&L EQUIP FULL SERVICE APPLICATION STOP IN 477 FT
SPEED - M.P.H.
DISTANCE - FEET

Graphic summary of the advantages the empty and load freight brake offers for grade service. The train equipped with the single capacity brake is running away on the 2.3 per cent. grade. The empty and load brake provides such a braking reserve that the train can be stopped from 20 m.p.h. in short of 500 feet on this grade.

pressure of 22.5 pounds—a very conservative assumption), and its reserve braking force enables it to stop from this speed in less than 500 feet.

In discussing car weights a newly designed freight car was mentioned of 75,000 pounds empty and 315,000 pounds loaded weight, which a prominent southeastern railroad is going to put into service in the near future. Only two years ago on this road the maximum tonnage per train was 6000 tons in a maximum train

length of 80 cars. These limiting conditions were determined by the ability of the single capacity brake to control trains over the critical district—" the neck of the bottle " for the system—which has a grade of 1.5 per cent. With the introduction of a limited number of empty and load brakes the tonnage was raised to 8000 tons per train. The difficulty now passed to the question of adequate length of sidings to care for the traffic volume, which was overtaxing the capacity of this single-track line. To provide the proper number of sidings of suitable length would be to virtually double-track the road. The use of the new 240,000-pound capacity cars will, however, render possible the operation of trains of 15,000 tons, and postpone thereby the double-tracking of this line for 25 years, according to the estimate of the railroad people. Full credit must be given to the empty and load brake for this saving, for *it would be absolutely impossible to operate such trains of these cars without the use of an empty and load brake.*

VOLUMES OF AIR.

Particular mention has been made of the air volume limit in the capacity of the single capacity brake to control modern freight trains. As equipment, both freight and passenger, has become heavier and trains longer the volume of air required for control of these trains has increased proportionately and the difficulty in supplying it in greater proportion. So it may be said that *the ability to run long trains depends almost entirely on the devices which permit the supply of air to reach the brakes on the rear end.* The difficulty always comes in releasing and recharging the brake system, for it is ever a simple matter to make the reduction necessary to apply the brakes. In other words, it is easy to allow pressure to escape, but always difficult to build it up.

In answer to the suggestion that a larger brake pipe be introduced, it can be said that, while a larger brake pipe would facilitate the transmission of air and utilize to good advantage increased compressor and main reservoir capacity on the locomotive, it would be impossible to increase the size of the brake pipe materially without disorganizing the whole air brake service, much as the introduction of the six-foot gauge would disrupt our transportation service in general. There could be no satisfactory interchange of equipment during the transition period, and the advantages of the enlarged brake pipe would not appear until practically every car

in the train was so equipped. To obtain adequate main reservoir capacity for the present side of brake pipe requires the utilization of every available space on the modern locomotive. To increase this capacity in order to realize the advantage of a larger brake pipe would be to present serious difficulties in the way of locomotive design. It is relevant to reconsider the 50 per cent. reduction in air consumption effected by the empty and load freight brake in this question of brake pipe capacity.

Modern passenger brake equipment is so designed that the pressure restored to the brake pipe serves to release the brakes only until the last stages of recharging are reached. With the old equipment, which served admirably for short trains of vehicles with small air requirements, the brake pipe recharged each brake immediately after releasing it. As train lengths and air consumption per vehicle became greater the head end equipments had more and more the tendency to bleed the brake pipe of air as fast as it was supplied, with the result that the head end was released and almost recharged before the rear end started to release. Modern brake equipments employ a supplementary storage reservoir, which is used for recharging the auxiliary during the release of the brake and for obtaining high cylinder pressure in emergency applications.

With the modern freight brake equipment (both single capacity and empty and load) the high pressure in the head end of the brake pipe, when releasing the brakes, causes the release of the head brakes to be retarded so that the rear brakes, starting later to release though they do, complete their release in practically the same time. This prevents the head cars from fully releasing ahead of and running away from the rear cars when a release of the brakes is made before the train is brought to a standstill. A retarded recharge for the head brakes permits the brake pipe air to pass on back to operate the rear brakes in a manner very essential to the successful operation of long freight trains. The importance of the ability to release and recharge the brake system promptly can only be appreciated by those who have had to contend with delay reports occasioned by control equipment used on trains of length and size beyond its capacity.

Where seconds are golden in the operation of trains, as on the New York subway system, the prompt release of a full application of the brakes on an entire train in five seconds with the new electro-pneumatic equipment, as compared with the 10 and even 17

seconds with preceding equipment, will prove to be a boon to the operating officials of that greatest railway system on earth. As retardation does not cease, or full acceleration take place, until all the air is exhausted from the brake cylinder, this saving of from 5 to 12 seconds in time, and even more when retardation is taking place, means much to snappy movement of trains. And it is well to mention that graduated release flexibility has not been impaired in this shortened time for a complete release.

TRAIN CONTROL FORMULÆ.

Computations with regard to train control have been made at various points in this paper, and a summary of all the formulæ is given in Fig. 15.

You will note that speed does not appear in the gradient formulæ. This is because no more retarding force is needed to hold a train down a grade at a speed of 40 miles per hour than at 10 miles per hour. But it is harder to get this retarding force at 40 miles per hour than it is at 10 miles per hour, because brake shoe friction is much less at the higher speed, due to the higher velocity of the wheel surfaces and the greater brake shoe heating. Therefore, the allowance for speed must appear in the value of coefficient of brake shoe friction used.

The average brake cylinder pressure needed to provide the necessary retarding force is the average between all minimum pressures occurring while the brakes are released (retainer pressures, if retainers are used) and all maximum pressures occurring while brakes are applied.

DRAFT GEAR CAPACITY.

It seems hardly necessary to dwell on the need for draft gears of improved shock-absorbing capacity when heavier cars, longer trains, locomotives of increased tractive effort, and larger velocity differences between cars in a train all point to shocks of ever-increasing intensity. It seems to be a very short-sighted policy to buy draft gears by the pound without regard for their operative characteristics and capacity—much more so, in fact, than the buying of fuel with no consideration for its calorific value.

When all trains are operated with multiple-unit control and the electro-pneumatic brake, the need that such careful attention be given to the draft gear problem will have passed away. Until then, however, it is deserving of careful consideration.

SUMMARY OF FACTORS GOVERNING TRAIN LENGTH.

We have seen that locomotive capacity, length of sidings, station platform lengths (for electric traction service), train control, and draft gear capacity determine and limit the number of cars that can be satisfactorily operated in one train.

FIG. 15.

TRAIN STOPS AND CONTROL ON GRADES.

FORMULAE.

$$S = 1.467\,Vt + \frac{WV^2}{30\left(\frac{p}{C}PW_e ef - GW\right)}. \quad (1)$$

$$= 1.467\,Vt + \frac{V^2}{30\,(Pef - G)}. \quad (2)$$

when $p = C$, & $W = W_e$.

$$= \frac{V^2}{30\,Pef}. \quad (3)$$

when $G = 0$ (level track) & "t" is neglected.

$$G = \frac{p\,PW_e ef + rTC}{CW}. \quad (4)$$

$$p = \frac{(GW - rT)C}{PW_e ef} \quad (5)$$

$$R = \frac{p\,PW_e ef}{CW} - G. \quad (6)$$

LEGEND:

R = retardation factor, percent

S = stop distance, feet

V = initial velocity, m.p.h.

G = gradient; + downhill. − uphill.

P = braking ratio, based on

C = basic cylinder pressure for C. Lbs. per sq. in.

p = average cylinder press. (for grade computation) = actual cylinder press. (for stop computation)

W_e = empty weight of car, basis for P and C.

W = actual weight, lbs.

T = actual weight, tons.

r = internal resistance of car, lbs. per ton.

e = efficiency of brake gear, averaging 85%.

f = coefficient of brake shoe friction, averaging from 10% to 20%.

t = average time, sec., train may be considered running free before the brakes get into action.

$P_L W_L$ may be substituted for PW_e when E&L brake is used.

P_L, W_L = loaded car braking ratio and weight.

The relation of train control to train length has been enlarged upon, and investigation reveals that the air brake art has kept abreast of—in fact, always in advance of—developments in all

the many other phases of railroad activity. It is well that it has been so, for the other developments would have been impossible of full realization otherwise. A concrete example is seen in the case where the empty and load freight brake is responsible for the postponement for 25 years of double-tracking a single-track line. The importance of reduced time of serial brake action has been made clear, it is hoped, and the part played in this by the development from the first type of automatic brake to those types employing the quick-service feature, and, finally, to the perfected electro-pneumatic brake, which has reduced the time element, and therefore the shock, to zero for trains of any length and for any rate of retardation up to the limit set by the adhesion between the wheel and rail, and which is a time-saver of the first order in the way of getting the brakes on and in getting them off. Increasing weights of vehicles and increasing velocity differences between these vehicles in the ever-lengthening trains of to-day make it imperative that modern types of brake apparatus be utilized to realize more fully the opportunities at hand for increasing the efficiency of railroad operation.

The importance of uniform release and uniform recharge features of brake apparatus and the 50 per cent. saving in air consumption provided by the empty and load brake has been dwelt upon in the matter of the capacity of the brake pipe, now at its maximum, for supplying air to all the vehicles in the train.

The extreme necessity has been emphasized of employing a foundation brake rigging which will permit the full realization of locomotive performance in capacity and economy, of the advantages of all the features embodied in modern air brake apparatus, and of smooth handling of long trains with a minimum of expense and delay.

In short, the best brake is the one which, in setting up the maximum retardation, creates the least velocity differences between vehicles in a train, and which has the greatest flexibility in providing *any* degree of retardation from the very minimum to the maximum.

TIME OF STATION STOPS.

Where traffic demands are continually treading on the heels of the supply, any saving in the time a train is held at a station to receive and discharge passengers will materially increase the train-mile performance of that train. Therefore, the car door

arrangement and capacity for ingress and egress of passengers on each car should be such as to permit the "flow" of passengers along "stream lines" of the greatest number and least interference one with the other.

The same is true of the station arrangement for dividing up this flow and directing it into the proper channels.

The local conditions as to the number of passengers to load and unload will, of course, make the time of station stop a variable factor between one point and another.

Where it is necessary to change the make-up of trains which are to continue in regular service, as in interurban and rapid transit operation, any device like the automatic car, air, and electric coupler, which will make or break all air and electric connections simultaneously with the coupling or uncoupling of cars, is a time-saver of the first order.

DELAYS TO TRAINS.

Obviously, if the yard capacity at division points is such as to require dispatchers to delay trains on the road pending the ability to receive them at the terminal, improvements made for improving road conditions can profit but little; and similarly with making up trains and starting them out.

The relation of defective or inadequate equipment in causing delays to trains needs but brief mention. The use of archaic equipment in the way of foundation brake rigging and air-controlling devices can but promote trouble in operating trains of modern weights and lengths. Delays due to stuck brakes, hot boxes, slid flat wheels (where passengers have to be transferred from one sleeper to another in the dead of night), hard pulling trains, and break-in-twos are some of the troubles which can readily be avoided by paying due attention to the underlying principles existing in the relation of train control to train operation.

ACCELERATION.

Traffic capacity will depend on the time needed to accelerate a train up to a maximum speed, and this, in turn, will depend upon the motive power, locomotives, steam and electric, and motor equipments for electric railways.

The accelerative performance of motor equipments depends on a host of things such as design, clearances, etc. Among these are

line-voltage regulation and the assistance here rendered by the current returned to the line due to regenerative braking in the control of trains on grades and in making stops. Selective relays operated by differences in the condition of car loading provide uniform acceleration, irrespective of whether the car be empty, partially or fully loaded.

The comparative number of motors and trailers in the train will, of course, determine the rate of acceleration possible. The new cars for the New York Municipal are all motors. The operation of all motor trains is the reasonable way to provide for maximum traffic, which goes without saying.

Much has been claimed, and deservedly, for the superheater, the brick arch, the feed-water heater, and the mechanical stoker in improving locomotive performance in the way of capacity and economy. The modern locomotive is really a very efficient machine, having a water rate as low as 14.5 pounds, compared with 8.5 for large steam turbine units. Except at very low speeds, locomotive capacity is entirely dependent upon the ability of the boiler to supply steam, and all of these improvements pertain directly to the boiler. In many cases the advantage of these improvements in the way of increased tonnage has been cut off by the inability to start the trains from rest. It is readily seen, therefore, that acceleration is governed by the brake shoe clearance maintained, for the harnessing of improved boiler performance cannot be realized if the train cannot be started under way. A foundation brake rigging which will maintain proper brake shoe clearances and thereby eliminate increased train resistances due to dragging brake shoes is vitally pertinent to each and every class of railway service.

Conditions of grade will obviously affect acceleration of trains. Ascending grades for the approach to stations and descending grades for the departure therefrom are of great importance and efficiency in improving rates of acceleration and retardation. These have been employed on the New York subway at a number of points.

MAXIMUM SPEED.

Acceleration is the first factor to dictate the maximum speed physically possible. The station spacing enters in the way of providing or failing to provide, first, the time necessary to get up to maximum speed, and, second, the period during which this speed

can be maintained before it is necessary to stop, which influences the average speed over a division.

The retardation phase, however, determines what is the safe maximum speed. New brake devices, which give for emergency high retardation rates over those developed for service operation of the brakes, and electro-pneumatic control, which makes these high rates possible without tearing up the train, have boosted the safe speed to a point which for other reasons railway managers do not care to approach.

In grade service the maximum speed will depend upon the reserve braking force, the heating of brake shoes and car wheels, and local conditions, such as curvature and changes in grade. With the single capacity brake it is necessary to crawl down mountain grades, because the margin, or breaking reserve, between control and a runaway train is so very small. At low speeds the coefficient of brake shoe friction is high, making a more efficient brake, and the time provided for recharging the brake system, before the increase in speed has taken the brake shoe friction to a greatly reduced value, is prolonged. The empty and load brake, with air consumption reduced one-half and brake effectiveness increased two and a half, or more, times, permits materially increased speeds down mountain grades. The braking reserve, increased anywhere from 500 to 1000 per cent. more, enables a stop to be made in a very short distance at any time during the descent of the grade, where the corresponding stop with the single capacity brake is measured in thousands, instead of hundreds, of feet.

Curves also have an influence on the maximum speed which can be made. Slow-downs for curves consume both time and energy.

RETARDATION.

The relation of retardation to the traffic volume handled by the train unit has been discussed at great length elsewhere in this paper. Retardation depends, of course, on the maximum speed, because the energy to be removed from a moving train in bringing it to rest varies as the square of the speed; that is, as the speed is doubled the stop distances will be multiplied four times, other things remaining constant.

The energy of a moving car varies directly with the mass or weight. Therefore, other things remaining equal, the stop dis-

tance will lengthen as the weight is increased. This explains why the hand brake is so inadequate for modern car weights. The brakeman of to-day is no stronger than the one of yesterday, but car weights have doubled, tripled, and even quadrupled. If more effectiveness is sought by increasing the leverage, or by applying hand brakes separately to each truck, the time element for getting the brake into action is greatly increased, which offsets the gain in braking effort.

Every second's delay in getting the brakes into action means, at a speed of 60 miles per hour, the addition of 88 feet to the stop distance. To handle the great volumes of air required for modern equipment with the least possible loss of time requires the use of devices of the most careful and scientific design.

The questions of brake shoe duty; uniform braking ratio; number of cars; serial action; air brake equipment necessary for cutting down the times for getting the brakes applied and for serial action, and for maintaining a constant braking ratio, whether the car be empty or loaded; a suitably designed foundation brake gear which will avoid, among many others, that evil in the form of low effective retardation due to the attempt to "dribble" on the brakes and avoid shocks—all of these have been examined before.

RAIL CONDITION.

When the brakes are applied on a car, each wheel thrusts forward on the rail with a force equal to the brake shoe friction on that wheel. The equal and opposite thrust of the rail against each wheel is the force which, applied from a point external to the vehicle, causes retardation. Obviously, if a demand of this sort is made on the rail in excess of the static or rolling friction (generally termed "adhesion") between the wheel and the rail, this friction will fail to keep the wheel turning and the brake shoe friction will cause the wheel to slide. The wheel-rail friction, now kinetic in nature, will be very much reduced and the actual retarding force acting on the wheel correspondingly reduced. From this it is evident that *the braking problem must begin and end with the rail.* If the condition of the rail surface is bad—which is another way of saying that the coefficient of friction is low—the amount of brake shoe friction that can be used without causing wheel sliding is very much reduced, and likewise the retardation possible.

As before mentioned, the retardation in percentage (or, what

is the same thing, the actual retarding force in relation to the weight of the vehicle) is expressed according to standard nomenclature.

$$R = \frac{p}{C} P\,ef$$

Where the actual cylinder pressure (p) equals the pressure (C) used as a basis for the braking ratio (P), this expression is simplified, thus:

$$R = P\,ef$$

Now if A be the designation for the adhesion, or coefficient of rolling friction between wheel and rail, the critical point for wheel sliding will be when the retarding force equals the adhesion; that is,

$$P\,ef = A$$

The actual value of the adhesion, A, will vary from 12 to 30 per cent., depending upon the weather conditions of temperature and relative humidity. With sand on the rail it may run even higher than 30 per cent. Taking an efficiency factor of 8 per cent. and an adhesion value of 25 per cent. (which is representative of the usual condition of rail surfaces), the braking ratio (P) necessary to slide wheels is

$$P = \frac{A}{ef} = \frac{.25}{.08} = 312.5 \text{ per cent.}$$

On the other hand, if the adhesion drops to 12 per cent., due to uncontrollable weather conditions, the braking ratio necessary to slide wheels is only

$$P = \frac{.12}{.08} = 150 \text{ per cent.}$$

If in the latter case the efficiency factor be 24 per cent., the braking ratio need be but 50 per cent. to cause wheel sliding. Thus it is appreciated that the dependence of the whole problem of braking comes not, as popularly believed, upon the question of braking ratio alone, but upon the values of wheel-rail and shoe-wheel friction as well for here we have illustrations of wheel sliding with braking ratios varying from 50 to 300 per cent.

REGENERATIVE BRAKING.

The difference between a train in motion and a train at rest is one of kinetic energy content. In order to bring a moving train

to rest it is necessary to remove this energy, and until recently the only available means was to cause it to flow from the train through the brake shoes in the form of heat energy and to be dissipated and lost in the surrounding atmosphere.

As illustrated in Fig. 2 (p. 427), the energy content of a modern train, due to greatly increased mass and velocity, is such that, could it be properly directed and harnessed, it would carry the full load of an 8000-kilowatt power plant for one minute's time. Any means, then, for saving this energy for use in accelerating trains is an economy of vital importance. Electric operation of trains provides an opportunity for effecting this saving in that suitable motor control apparatus can direct the driving effect of the moving train to operate the motors as generators and return thereby the kinetic energy of the train to the line in the shape of electrical energy. This is regenerative braking. In addition to the energy saving, wear and tear on car wheels and brake shoes is avoided.

However, contrary to the expectations of the uninitiated, the need for modern air brake installations is just as pressing with regenerative braking as without it, for otherwise, in event of any failure in the line or in the motor equipment, the train would be altogether uncontrolled. Moreover, where regeneration is employed with the electric locomotive, the responsibility for control is vested in one or two units, and a failure of one or both means a failure of half or all the power to control. On the other hand, with an air brake equipment on every car in the train, a failure of one, two, six, or ten units (depending on the total number in the train) will be of relative insignificance.

Thus it is apparent that to realize the best economies in the control of freight trains down mountain grades with the regenerative brake it will be necessary to employ the empty and load brake in order to provide the braking reserve indispensable to speed and safety. Otherwise, if the single capacity brake is employed and the safe speed for this type of brake is exceeded at any time during regeneration, a failure of the regenerative brake means a runaway train.

The subject of regenerative braking is worthy of a paper in itself, but the foregoing sketch will suffice for the purposes of this general treatise.

SYSTEM OF TRAINS.

HEADWAY OR SPACING OF TRAINS.

The fundamental consideration for the headway or spacing of trains is the element of *safety*. Safety of operation in turn depends upon the possible retardation or ability to stop; the maximum speed; and the installation and characteristics of signal apparatus. These factors are arranged in order of relative importance, though, of course, each is closely bound up in, and not to be dissociated from, the others.

The minimum headway for the movement of trains may be determined in two different ways. One method is to base the proper time interval between trains on a system of trains *running at maximum speed*. The other is to base this time spacing on the *closing-up of trains at stations*. The method of these two which gives the larger minimum headway must be the one to use for the conditions in question, for, obviously, of two critical values the safer, which is the larger, must always be chosen.

In the subsequent analysis certain assumptions are to be understood, namely: straight and level track with no irregular local conditions and carrying traffic in one direction only; a block section equal in length to *twice* the emergency stop distance from the maximum speed; stop distances proportional to the square of the maximum speed; duration of retardation directly proportional to the maximum speed.

It is needless to attempt an investigation of this kind with a thousand-and-one variables. The modifications necessary to apply the results herein established to special conditions of grade, curvature, interlockings, and other local conditions will be apparent, once these results are understood. The above assumptions are sufficiently accurate in every respect to make worth while this analysis of the factors influencing headway for train movements.

No attempt will be made to deal with the laying out of a signal system for the system of trains here considered. These data can be found elsewhere, a couple of very good treatments of this proposition being found in Mr. H. G. Brown's paper, "The Signalling of a Rapid Transit Railway," read before the Institution of Electrical Engineers in April, 1914, and in Mr. M. L. Patterson's article, "Methods of Locating Automatic Signals," in the *Railway Signal Engineer* for February, 1916.

In general, trains running at speed should be spaced by a distance equal to the sum of:

1. The length of the train.
2. The length of the complete block section.
3. The distance between the distant and home signals.
4. The distance sufficient to permit the signal to clear and the engineman to identify the signal indication.

In the New York subway the overlap system of signals is used, which provides for *two* home signals and one distant signal protecting the rear of each train. In order to give each following

FIG. 16.

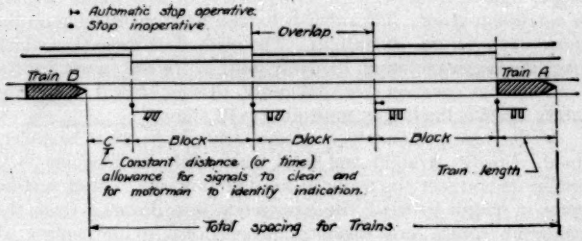


DIAGRAM OF OVERLAP SYSTEM OF BLOCK SIGNALS, ILLUSTRATING SPACING OF TRAINS.

train a clear distant signal (and this should be the normal condition), the complete block section plus the distance between the distant and the second home signal should be taken as three times the length of a block, or as six times the emergency stop distance.

The distance spacing for this system is illustrated in Fig. 16. Constant C (item 4 in the list above) may be a given *distance* or a given *time*. If taken as a constant distance, the time will vary, of course, according to the speed. The *time* spacing or headway between trains will be the time necessary to run this total distance spacing, or, expressed mathematically,

$$(1) \qquad H_R = \frac{6S_B + NL + C_S}{1.467\ V}$$

where:

H_R = headway determined by running at speed (seconds);
S_E = emergency stop distance (feet) from
V = maximum speed (mph);
C_S = *space* constant (feet);
N = number of cars in train;
L = length of each car (feet).

While the kinetic energy of a train varies directly with the square of the speed, the retarding force, due to the brake shoe friction, is decreased with an increase in speed. On the other hand, the initial or reflex time required for getting the brakes into action—constant, of course, for any speed—is of decreasing importance, relative to the total time for stopping, as the speed is increased. It is found that the influences of these two factors, reflex time and brake shoe friction, are approximately counteractive, and, therefore, the stop distance will vary directly with the square of the speed; that is, the braking distance will equal some constant times the square of the speed. Whence, substituting in (1)

$$H_R = \frac{kV^2 + NL + C_S}{1.467\ V} \tag{2}$$

If the allowance C, as illustrated in Fig. 16, be taken as a *time* instead of a space constant to permit signals to clear and motorman to identify them, (2) becomes

$$H_R = \frac{kV^2 + NL}{1.467\ V} + C_t \tag{3}$$

Solving (3) for V, we have in quadratic form

$$V = .733\frac{(H_R - C_t)}{k} \pm \frac{\sqrt{2.15(H_R - C_t)^2 - 4kNL}}{2k} \tag{4}$$

The curves for the relation between H_R and V as expressed by (2) are plotted in Figs. 18 and 19 and those for (3) in Fig. 21.

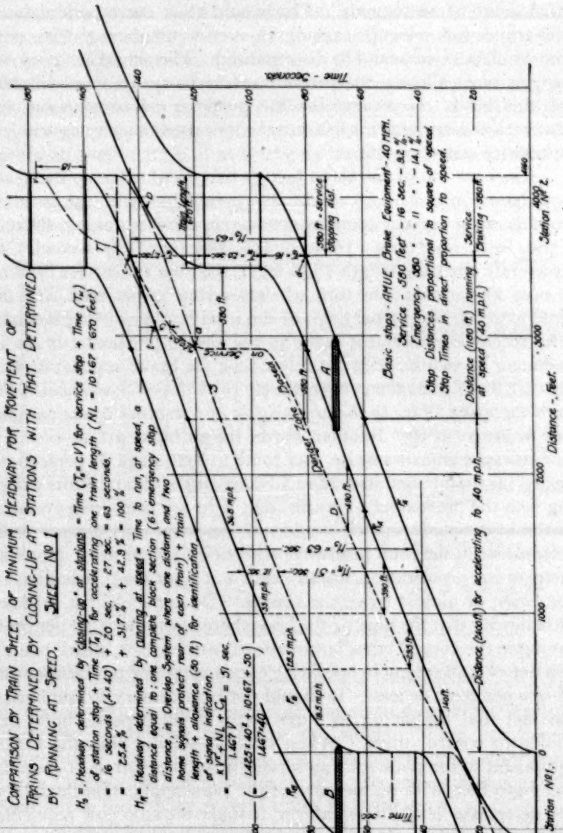
Curves for train lengths of zero, 335, and 670 feet are shown, and for each there is a critical speed for the minimum headway; that is, for each length of train there is a critical speed above or below which a headway obtains greater than that at the critical speed. A curve giving the locus of all such critical points for every length of train is indicated on both Figs. 18 and 21. The headway increases for speeds lower than these critical points because the length of train is of increasing relative importance; that

is, the length of train, with decreasing speeds, more nearly approaches the length of the block section—it being kept in mind that the latter is always a function of the speed under these considerations. The headway increases for speeds higher than the critical value because the braking distances and, therefore, the block sections are longer, increasing as the square of the speed, while the time required for covering a certain distance decreases in inverse ratio only directly as the speed.

The headway determined by closing-up at stations is illustrated in Fig. 17, and a comparison made with the headway determined by running at speed. The progress of two trains, B following A, from one station to another is traced by the lines F and R. These lines mark the progress of the front and rear ends of each train on a time-distance basis. Train B is shown just having left the first station at the time (70 seconds) after train A has advanced from this station about 3000 feet and is running at full speed. Train A stops at the second station, 4160 feet from the first, at about 96 seconds after leaving the first. After a station stop of 20 seconds as shown, it starts again for the third station (not shown).

Curve D (dotted) marks the danger zone inside which the head end of train B must not come if the rear end of train A is to be safe. This zone is based upon the service braking distances for the speeds at which train B is running under normal operation at the particular points in question along the right-of-way. Curve F_B is tangent to curve D at point c where braking must begin for the stop of train B at station two. The time interval between this critical point for train B and the corresponding point d for train A is the headway (H_S) determined by trains closing up at stations, and is seen to be the sum of: (1) the time to make the service stop (T_s); (2) the time the train is held at the station (T_w); and (3) the time required to accelerate one train length (T_a). This is the limiting or critical value for the headway and cannot, of course, be realized in actual practice, because, among other things, the time of station stop will change with the varying number of passengers to be handled. Suppose, under the condition shown, the time of station stop for train A had been lengthened five seconds. This would cause an intersection of curve D with F_B at t, requiring that train B start braking at this point and advance to the usual station stop at point v, as indicated by curve

FIG. 17.

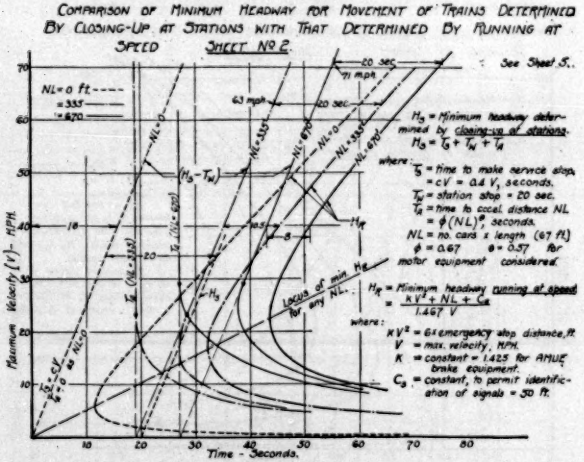

COMPARISON BY TRAIN SHEET OF MINIMUM HEADWAY FOR MOVEMENT OF TRAINS DETERMINED BY CLOSING-UP AT STATIONS WITH THAT DETERMINED BY RUNNING AT SPEED. SHEET No 1
Station No 1.
Station No 2.
Distance - Feet.
Time - Seconds.
Distance (each) for accelerating to 40 m.p.h.
Distance (less 1/2) running at speed (40 m.p.h.)
Service Braking - 350 ft.
Basic stops - AMUE Brake Equipment - 40 M.P.H.
Service - 550 feet - 16 sec. - 82 1/2 %
Emergency - 350 " - 11 " - 141 1/2 %
Stop Distances proportional to square of speed.
Stop Times in direct proportion to speed.
Service stopping dist.
Danger Signal
Danger Signal
Home Signal
Home Signal
Home

T. In this case the initial delay of five seconds has resulted in a final delay of 18 seconds. This would react correspondingly on the trains following *B*, causing an ever-accumulating delay until the whole service would be disorganized. The actual headway for service must allow a comfortable safety factor to avoid troubles of this kind. However, for the purposes of comparison, the theoretical minimum for headway will be used in every case unless otherwise noted.

Examination of the three factors that go to make up the headway based on closing-up at stations reveals that, with the assumed conditions of speed (40 miles per hour), service braking distance (580 feet) and time (16 seconds), and time (27 seconds) to accelerate one train length (670 feet), the time for service braking is only 25 per cent., the time of station stop 32 per cent., and the time to accelerate 43 per cent. of the total headway of 63 seconds. The recent improvement made in the electro-pneumatic brake in reducing by two seconds the reflex time for brake application still further diminishes the *comparatively slight interference which the modern brake offers to the proposition of operating trains without any headway at all*. In other words, the air brake art, as exemplified in rapid transit service, has come to that stage of perfection, has so far maintained its lead in advance of the other factors entering into the movement of trains, that a 10 per cent. improvement in the brake performance would now result in only 2.5 per cent. betterment in the total headway; whereas, a 10 per cent. improvement in the acceleration would mean a 4.3 per cent. reduction in headway, or almost twice the saving. Do not make the mistake of believing that the part of the brake here resembles a 10 per cent. saving in the cost of brass buttons for the conductor, which results in a net overall saving in operating expenses of one ten-thousandth of one per cent. or less. It should not be necessary to emphasize the fact that the ability to move trains at all depends upon the ability to control them. When it is considered that the brake equipment in the same subway service in 1906 required 41 seconds and 1450 feet, with no measure of the same application flexibility and no release flexibility whatever, to make the stop now requiring but 14 seconds and 500 feet, the significance may be better grasped of the above comparison as to realization of opportunity and fulfilment of economic trust in the development progress of the air brake.

Curve S, Fig. 17, represents the time-distance path of the front end of train B following A on a headway (H_R) of 51 seconds, determined by running at the maximum speed of 40 miles per hour. This is 12 seconds less than the headway (H_S) determined by closing-up at stations. Therefore the latter must govern; for note that curve S enters the danger zone at a, and if train B were permitted to continue as indicated by S, it would come within 40 feet of train A at b. Were anything to detain train A

FIG. 18.


COMPARISON OF MINIMUM HEADWAY FOR MOVEMENT OF TRAINS DETERMINED BY CLOSING-UP AT STATIONS WITH THAT DETERMINED BY RUNNING AT SPEED
SHEET Nº 2.
See Sheet 3.
Maximum Velocity (V) - MPH
Time - Seconds.
NL=0 ft
=335
=670
71 mph
63 mph
20 sec
H_S = Minimum headway determined by closing-up at stations.
$H_S = T_S + T_M + T_A$
where:
T_S = time to make service stop. $= cV = 0.4\ V$, seconds.
T_M = station stop = 20 sec.
T_A = time to accel. distance NL $= \phi(NL)^{\theta}$, seconds.
NL = no. cars x length (67 ft)
$\phi = 0.67$ $\theta = 0.57$ for motor equipment considered.
H_R = Minimum headway running at speed $= \frac{KV^2 + NL + C_a}{1.467\ V}$
where:
KV^2 = 6½ emergency stop distance, ft.
V = max. velocity, MPH.
K = constant = 1.425 for AMUE brake equipment.
C_a = constant, to permit identification of signals = 30 ft.
Locus of min H_R for any NL.

A detailed analysis and comparison of the two methods of figuring headway; one based upon "closing up" at stations, and the other upon "running at speed."

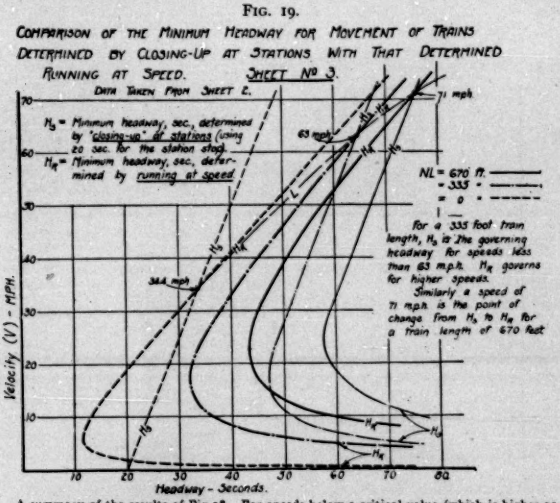
by one second or more, either at the station or after starting therefrom, a collision would be the result. Were braking to commence on train B at point a, due to a proper signal observance of the danger zone, the initial delay of 12 seconds would be multiplied into a final delay several times as great.

A more detailed analysis of the relations between the headways H_S and H_R may be found in Figs. 18, 19, and 20. Fig. 18 gives a complete development of the various factors which go to make up the two species of headway. The time for accelerating

different train lengths is given by the formula in the legend, and also graphically on Fig. 26. Some of the bases in the way of braking distances and times appear on Fig. 17.

Fig. 19 is a summary of results taken from Fig. 18. The heavy lines represent the "running" headway (H_R), and the light lines the "station" headway (H_S). The dotted lines are for zero train length; the dash and dot lines for a train length of 335 feet; and the full lines for a train 670 feet long. Curves for

FIG. 19.


COMPARISON OF THE MINIMUM HEADWAY FOR MOVEMENT OF TRAINS
DETERMINED BY CLOSING-UP AT STATIONS WITH THAT DETERMINED
RUNNING AT SPEED. SHEET Nº 3.
DATA TAKEN FROM SHEET 2.
H_S = Minimum headway, sec., determined by "closing-up" at stations (using 20 sec. for the station stop).
H_R = Minimum headway, sec., determined by running at speed
NL = 670 ft.
= 335 "
= 0 "
for a 335 foot train length, H_S is the governing headway for speeds less than 63 m.p.h. H_R governs for higher speeds.
Similarly a speed of 71 m.p.h. is the point of change from H_S to H_R for a train length of 670 feet
71 mph
63 mph
38.4 mph
Velocity (V) - M.P.H.
Headway - Seconds.

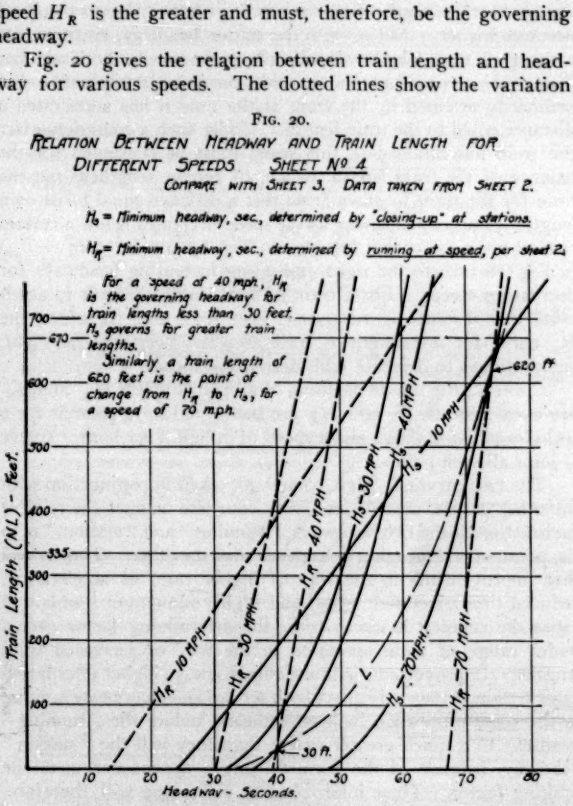
A summary of the results of Fig. 18. For speeds below a critical value (which is higher as the train length is increased) the headway based upon "closing-up" at stations will be the larger and therefore the determining value for the spacing of trains.

zero train length are given to show what the limiting conditions are when a train is considered as a point only. Curve L gives all points where, for any train length, H_R equals H_S; that is, it relates the speeds and these equal headways for any train length. Thus for a 335-foot train length H_S is greater than H_R, and will, therefore, be the governing headway for all speeds under 63 miles per hour. At that speed the two headway curves cross and mutually indicate a headway of about 64 seconds. Above this

speed H_R is the greater and must, therefore, be the governing headway.

Fig. 20 gives the relation between train length and headway for various speeds. The dotted lines show the variation

FIG. 20.


RELATION BETWEEN HEADWAY AND TRAIN LENGTH FOR DIFFERENT SPEEDS SHEET No 4
COMPARE WITH SHEET 3. DATA TAKEN FROM SHEET 2.
H_S = Minimum headway, sec., determined by "closing-up" at stations.
H_R = Minimum headway, sec., determined by running at speed, per sheet 2.
For a speed of 40 mph., H_R is the governing headway for train lengths less than 330 feet. H_S governs for greater train lengths.
Similarly a train length of 620 feet is the point of change from H_R to H_S, for a speed of 70 m.p.h.
Train Length (NL) - Feet.
Headway - Seconds.
700
670
600
500
400
335
300
200
100
10
20
30
40
50
60
70
80
620 ft
30 ft.
H_R - 10 MPH
H_R - 30 MPH
H_R - 40 MPH
H_R - 70 MPH
H_S - 10 MPH
H_S - 30 MPH
H_S - 40 MPH
H_S - 70 MPH
L

The running headway will determine the spacing of trains for train lengths less than a certain critical value (which is greater as the speed is higher).

of running headway with train length, and, similarly, the full lines for station headway. As the speed is higher the dotted lines become more and more erect, showing that the train length becomes of decreasing relative importance for the higher speeds; that is, a

change in train length affects the running headway less as the speed becomes higher. Not so with the station headway, however, for a change in train length naturally affects the time to accelerate that distance by a constant amount, unless the speed be lower than that ordinarily attained by the train at the time it has accelerated a distance equal to the train length. Under such a condition, after the train has attained it, this speed must be continued for the balance of the train length, which, of course, lengthens out the time for the train to move from rest a distance equal to its own length. This explains why as the speed decreases below a certain critical point the curves for H_S in Fig. 19 and for $(H_S - T_W)$ in Fig. 18 turn to the right, indicating increasing headways for decreasing speeds. For a train of zero length the time to accelerate that distance is, of course, also zero, and, therefore, the H_S curve for this length of train does not swerve to the right, but continues to decrease with decreasing speeds.

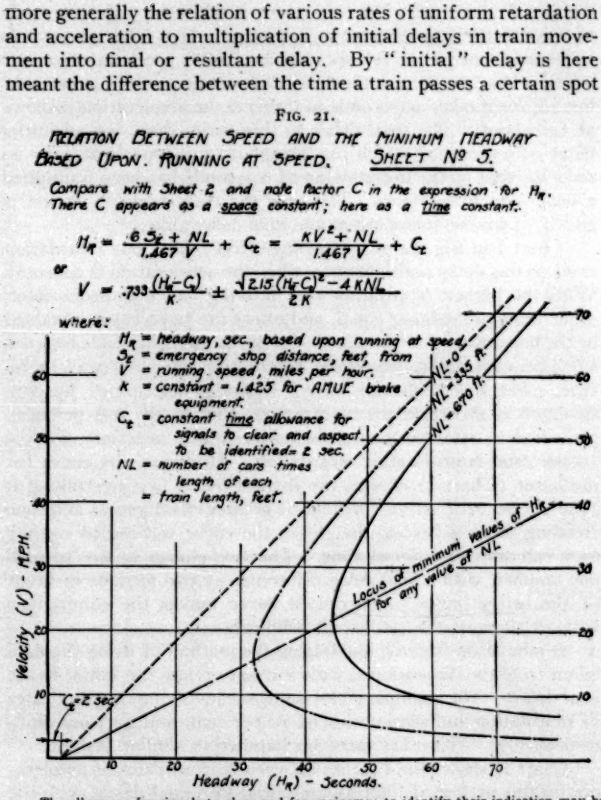
Curve L, Fig. 20, is the locus of all points where H_S and H_R are equal; that is, H_S and H_R are both equal to 74 seconds for a train length of 620 feet and a speed of 70 miles per hour. Curve L joins all such points.

The two curves L, Figs. 19 and 20, taken in conjunction with the other curves, describe in a very complete manner the relative jurisdiction of the two headways, "running" and "station," as to the point where one takes precedence over the other. They reveal that improvements in the way of higher rates of acceleration, reduced time of station stops, and higher maximum speeds will cause the running headway to be the determining factor over a wider range of train operation in the way of increased train lengths. Improvements in brake effectiveness (higher retardation rates), though they may affect both service and emergency braking in the same proportion, will nevertheless reduce the "running" headway to a much greater extent than they will the "station" headway, because of the former's larger dependence upon the braking factor. These improvements in braking will, therefore, have an effect opposite to the above by making the station headway the determining factor over a wider range of train operation in the way of increased speeds and longer trains.

An isolated example appeared in connection with Fig. 17, where an initial delay of 5 seconds to a train was multiplied into a final delay of 18 seconds. Fig. 22 has been prepared to portray

more generally the relation of various rates of uniform retardation and acceleration to multiplication of initial delays in train movement into final or resultant delay. By "initial" delay is here meant the difference between the time a train passes a certain spot

FIG. 21.


RELATION BETWEEN SPEED AND THE MINIMUM HEADWAY BASED UPON RUNNING AT SPEED. SHEET No 5.
Compare with Sheet 2 and note factor C in the expression for H_R. There C appears as a space constant; here as a time constant.
$H_R = \frac{.6\,S_t + NL}{1.467\,V} + C_t = \frac{kV^2 + NL}{1.467\,V} + C_t$
or
$V = .733\frac{(H_R - C_t)}{k} \pm \frac{\sqrt{2.15(H_R - C_t)^2 - 4kNL}}{2k}$
where:
H_R = headway, sec., based upon running at speed.
S_t = emergency stop distance, feet, from
V = running speed, miles per hour.
k = constant = 1.425 for AMUE brake equipment.
C_t = constant time allowance for signals to clear and aspect to be identified = 2 sec.
NL = number of cars times length of each = train length, feet.
NL=0
NL=345 ft.
NL=670 ft.
Locus of minimum values of H_R for any value of NL
C_t=2 sec.
Velocity (V) - M.P.H.
Headway (H_R) - Seconds.

The allowance for signals to clear and for enginemen to identify their indication may be taken in terms of distance or of time. This shows the result where the allowance is made in *time*. Fig. 18 makes the allowance in *distance*. This applies only to "running" headway, of course.

at the lowest speed to which it has had to slow down and the time it would have passed this spot had it continued at maximum speed. Thus, if a complete stop is made from 40 miles per hour in 18 seconds (a retardation rate of about 10 per cent.), the initial time

lost is not 18 seconds, but only 9, for it takes the train 9 seconds to traverse the stop distance at full speed. If it takes 40 seconds to return from rest to full speed at a uniform acceleration of one mile per hour per second, the delay this occasions is not 40 seconds, but 20, for it takes 20 seconds to traverse the acceleration distance at full speed. The total delay to this train, then, for an initial delay of 9 seconds (which just brings it to a stop) is 20 + 9 = 29 seconds; that is, the initial delay of 9 seconds has been multiplied a little over three times. The time the train remains at rest is added, of course, to the initial and final delay alike.

Chart I in Fig. 22 compares the effect of various retardation rates on this delay multiplication when the acceleration is constant. With the higher retardation rate braking can commence later, other things remaining equal, and effect the same result obtained by the lower rate of retardation. In the comparison made between a 6⅔ per cent. and a 10 per cent. retardation, the latter was, therefore, given an advantage of 1.8 seconds, determined by construction as shown in the main figure; that is, the 6⅔ per cent. train had to retard 1.8 seconds before it was necessary for the 10 per cent. train to start retardation. However, the curve for the latter (Chart I) crosses the former twice, first overtaking it (the 6⅔ per cent. curve) because of greater effectiveness and then receding from it beyond the jog in the curve because of coming to a full stop in reduced time. The two curves finally parallel one another, with a final delay difference of two seconds in favor of the better brake. The dotted curve makes the comparison without allowing the 1.8-second handicap.

A tabulation is given illustrating the method of using the data taken from the dimensioned main curve to relate the initial to the final delay. The example given is the one involving constant rates of retardation and acceleration of 10 per cent. and 4.55 per cent., respectively. The other rates are handled in similar fashion.

Chart II shows the relation between various rates of acceleration in this matter of multiplying initial into final delay, with the rate of retardation constant at 10 per cent., in order to have but one variable in the problem at a time. Up to the point of coming to a complete stop (indicated by the jog or turn in each curve) the 10 per cent. acceleration rate gives a multiplication factor of two, the 4.55 per cent. (one mile per hour per second) a factor of 3.3, and the 2.25 per cent. rate multiplies the initial delay by 5.9 into

FIG. 22.

RELATION OF VARIOUS RATES OF RETARDATION AND ACCELERATION TO MULTIPLICATION OF INITIAL DELAY IN TRAIN MOVEMENT INTO FINAL OR RESULTANT DELAY.
(INITIAL AND FINAL SPEED 40 MPH)

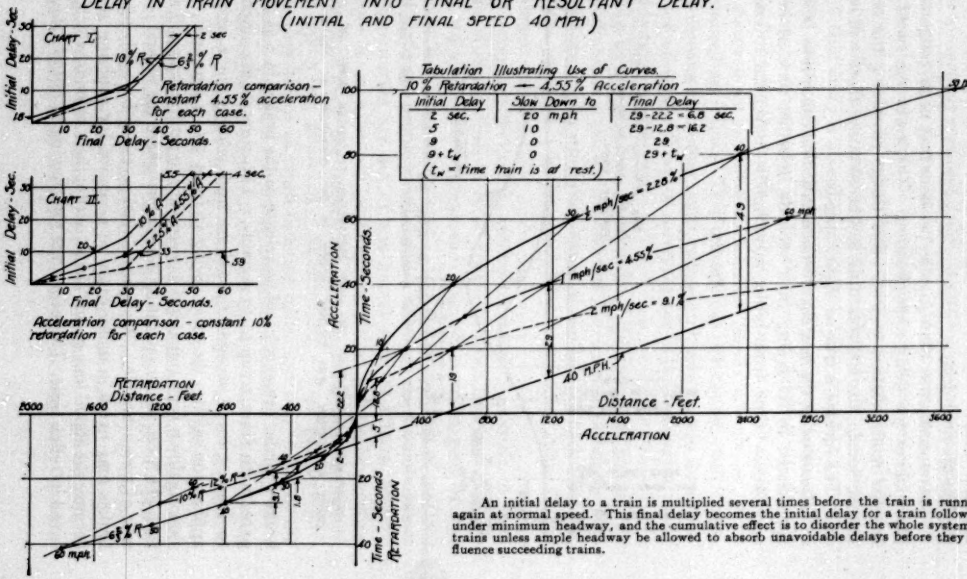


CHART I.
Initial Delay - Sec.
Final Delay - Seconds.
2 sec
10% R
6⅔% R
Retardation comparison - constant 4.55% acceleration for each case.
CHART II.
4 sec.
Acceleration comparison - constant 10% retardation for each case.
Tabulation Illustrating Use of Curves.
10% Retardation — 4.55% Acceleration
Initial Delay | Slow Down to | Final Delay
2 sec. | 20 mph | 29 - 22.2 = 6.8 sec.
5 | 10 | 29 - 12.9 = 16.2
9 | 0 | 29
9 + t_w | 0 | 29 + t_w
(t_w = time train is at rest.)
Time - Seconds.
ACCELERATION
RETARDATION
Distance - Feet.
20 MPH
40 MPH
60 MPH
80 MPH
½ mph/sec = 2.2%
1 mph/sec = 4.55%
2 mph/sec = 9.1%
10% R
6⅔% R
4.5
1.8

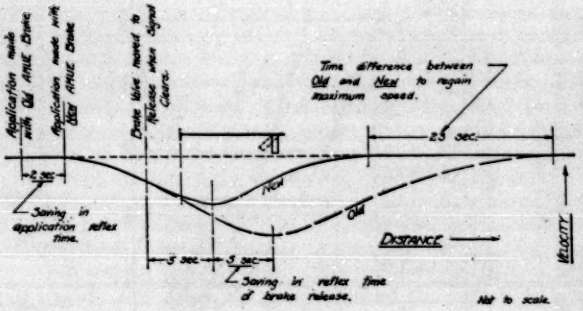
An initial delay to a train is multiplied several times before the train is running again at normal speed. This final delay becomes the initial delay for a train following under minimum headway, and the cumulative effect is to disorder the whole system of trains unless ample headway be allowed to absorb unavoidable delays before they influence succeeding trains.

the final delay. As is to be expected, the better the acceleration the less the final delay.

Any speeds other than 40 miles per hour may be investigated in like manner for delay multiplication. The lower the speed the less the multiplication will be, other things remaining the same, for, obviously, the less retarding and accelerating there will be to do. The reverse is true of higher speeds.

The value of the new electro-pneumatic brake may now be well appreciated in its saving of 2 seconds in reflex time of brake application (time from brake valve handle movement to rise in

FIG. 23.



GRAPHICAL ILLUSTRATION OF WHAT THE SAVING IN REFLEX TIME OF BRAKE APPLICATION AND RELEASE MEANS TO THE OPERATION OF TRAINS

brake cylinder pressure) and its release of the brakes in 5 as compared with from 10 up to 17 seconds. This time saving amounts to from 5 to 12 seconds for release and, adding the 2 seconds application saving, the total saving is from 7 to 14 seconds. In slowing down for a signal it is as necessary to get the brake off as it is to get it on, under operating conditions where the second is the unit of time schedule. And the delayed necessity for applying the brakes may mean in many cases no necessity for applying at all, because the signal may go to "clear" within the 2 seconds which are saved in reflex time. Therefore, the total saving in *initial delay of*

from 7 to 14 seconds means a final total saving of anywhere from *14 to 40 seconds resultant delay*, depending on the multiplication factor applying in the particular case. If the minimum saving of 7 seconds be taken with the acceleration rate of one mile per hour per second, which is higher than actually realized, and with a 10 per cent. retardation rate, according to Chart II, Fig. 22, the saving in final delay is about 23 seconds. A 23-second initial delay for the following train is multiplied into 43 seconds final delay, and this, as initial delay for the third train, becomes 63 seconds initial delay for the fourth train, and so on. The significance of the 7-second saving becomes apparent in its cumulative effect on succeeding trains. This presupposes, of course, that trains are following each other under minimum headway, and the necessity for allowing a delay, or safety, factor in the headway actually used is again emphasized.

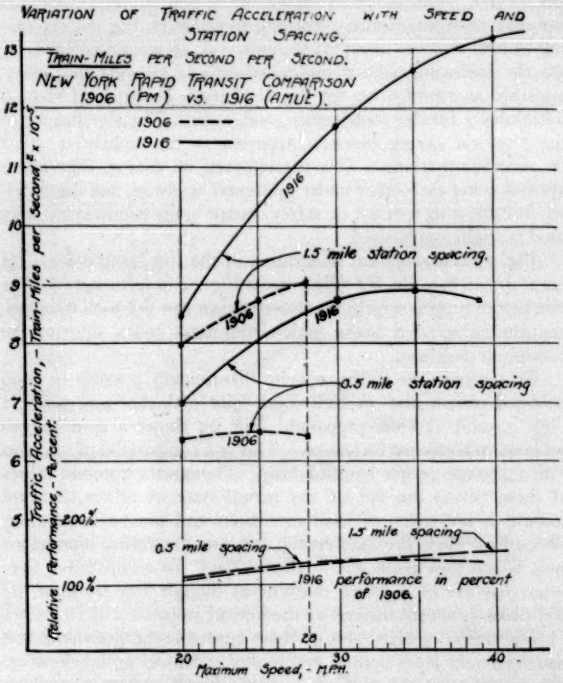
Fig. 23 is a graphical illustration of the foregoing points. It is not drawn to scale, but supplements Fig. 22 in picturing the time lost before a train which has slowed down can get back to speed, and the value of a brake which eliminates reflex time to the maximum degree.

To compare the traffic capacity theoretically possible in 1906 on rapid transit lines in New York City with that now possible Figs. 24 and 25 were prepared. Fig. 25 shows a gain of 350 per cent. in traffic performance. This is a comparison of 300,000 with 1,350,000 people handled daily. Though the absolute values of these curves are not of any actual state of affairs, because account is not taken of local conditions and practical operating allowances, they are interesting for the theoretical maximum limit which they point out. Nevertheless, for comparative purposes they are of as much meaning as though they were, in the first place, practical instead of theoretical values.

The traffic capacity unit of these curves is the *train-mile (or passenger-mile) per second per second*. This is an acceleration unit representing the *increase in traffic handling facilities which can be made in unit time* to care for the peak loads of rush hours. This acceleration unit is found by dividing the average or schedule speed over a given distance (including station stops) by the headway. If the average speed, V_s, be in miles per second and the headway, H_s, be in seconds per train, the quotient will be "train-miles per second per second." The corresponding unit, "passen-

ger-miles per second per second," is found by multiplying the train-mile factor by the train capacity in passengers. This factor stands in the same relation to train-miles and time that accelera-

FIG. 24.



The average or schedule speed of trains in miles per second over a district divided by the seconds headway intervening gives a *traffic acceleration unit*—train miles per second per second. This is a measure of the ability to get trains into operation quickly to care for peak traffic loads. Modern brake equipment has rendered possible an increase of 50 per cent. over 1906 performance in the number of longer trains it is possible to introduce into operation in a certain period of time.

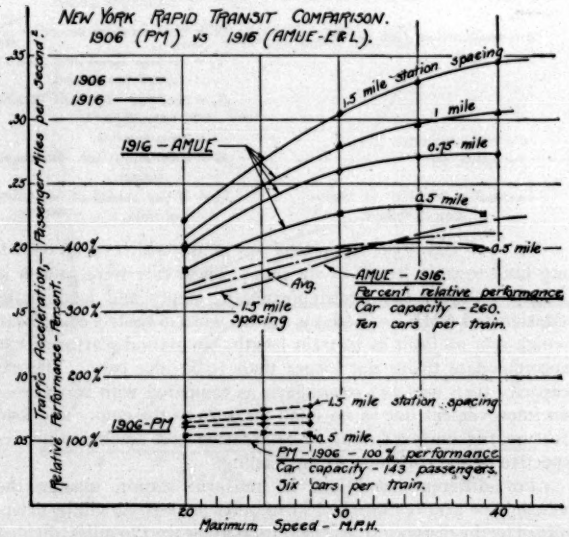
tion, as generally known, stands to space and time. Thus, where velocity equals acceleration times the time, the number of train-miles per second operating at any instant equals the traffic accelera-

tion factor multiplied by the time during which it has been in play. Of course, after a railway system has become completely filled with trains a maximum velocity, or number of train-miles per second, has been attained and more trains cannot be introduced; also, as

FIG. 25.

VARIATION OF TRAFFIC ACCELERATION WITH SPEED AND STATION SPACING.

PASSENGER-MILES PER SECOND PER SECOND.



The results of Fig. 24 are here shown in terms of *passenger-miles* per second per second. Due to the use of modern train control equipment, it is possible to introduce 350 per cent. more traffic-handling facilities in a certain period of time than was possible in 1906.

the number of points for introducing trains is increased the portion of the entire railway system per point is decreased and, therefore, the time for filling to maximum capacity is also decreased. In this discussion it is to be understood that the term "railway

system" applies to that portion supplied from *one point*. Many interesting analogies will come to mind in the consideration of this newly evolved acceleration factor, of which extended discussion will not be here made. Mathematically these analogies may be summed up:

SPACE	TRAFFIC
1. $v = v_o + at$	1. $V = V_o + At$
2. $s = v_o t + \frac{1}{2} at^2$	2. $M = V_o t + \frac{1}{2} At^2$
3. $v = v_o + \sqrt{2as}$	3. $V = V_o + \sqrt{2AM}$
where:	where:
v = velocity, feet per second.	V = train-miles (or passenger-miles) per second.
a = acceleration, feet per second^2.	A = train-miles per second^2 = $\frac{V_s}{H_s}$
	V_s = average speed over district, miles per second.
	H_s = headway between trains seconds.
t = time, seconds.	t = time, seconds.
s = space. feet.	M = train-miles (or passenger-miles).
v_o = initial velocity at beginning of time, t.	V_o = M per second at beginning of time, t.

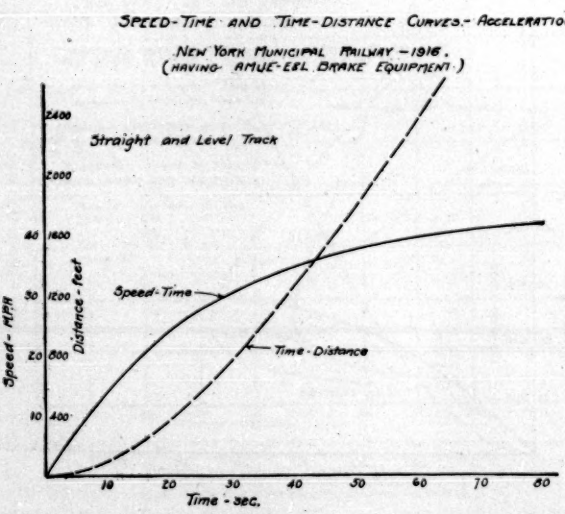
In 1906 PM brake equipment was in use, which set an operating limit to train length of six cars. These cars were 51 feet in length. To-day the electro-pneumatic, empty and load, brake (designated AMUE – E&L) is the last word in control equipment, which sets no limit as to train length, but station platforms can accommodate trains not longer than 10 67-foot cars. The car capacity then was 143 passengers, as compared with 260 now—an improvement due in no small measure to the empty and load feature for uniform rates of retardation and acceleration irrespective of the condition of car loading.

For different conditions of uniform station spacing the schedule or average speed (V_s) in miles per second will be determined by the distance from one station to the next in miles, divided by the sum, in seconds, of the times of: acceleration, running at maximum speed, deceleration, and station stop. Figs. 26 and 27 give, for the old and new conditions, the acceleration characteristics used as bases in the comparative traffic capacity curves of Figs. 24 and 25.

The basic times and distances for braking are, for the 1906

(PM) equipment: 650-foot emergency stop in 21 seconds; service stop in 1470 feet and 40 seconds—both from 40 miles per hour. As seen from Fig. 27, the maximum speed does not exceed 28 miles per hour. The corresponding stops from 40 miles per hour for the 1916 (AMUE) equipment are: 380 feet in 11 seconds for emergency, and 580 feet in 16 seconds for service. For all other speeds the braking distances are assumed to vary as the squares of the speeds and the times directly with the speed. The time of

FIG. 26.

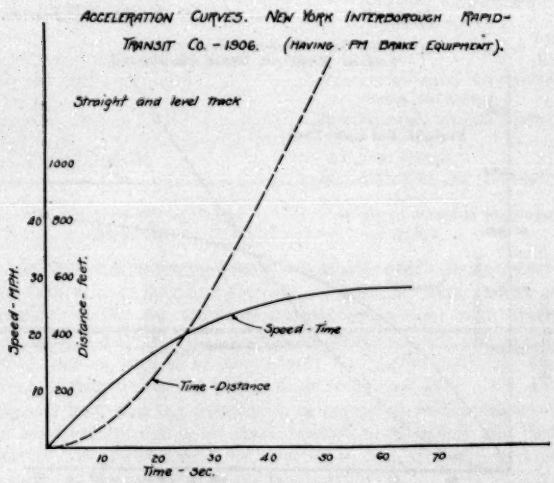


station stop is taken in every case as 20 seconds. The "station" headways (H_s) of Fig. 19 are the values used for the modern performance shown by Figs. 24 and 25. Headways similarly determined are used for the curves of 1906 train operation.

Fig. 28 compares the "running" and "station" headways for modern equipment with those of 1906. The train length for each case is approximately the same, which eliminates any uncertainty due to the influence of this factor. The minimum "running"

headway required for modern operation is only from 63 per cent. (at 60 miles per hour) to 65 per cent. (at 30 miles per hour) of that required for 1906 operation. The minimum "station" headway required varies from 63 to 72 per cent. in like manner. All of this may be summed up by saying that modern train control equipment has cut down the minimum headway required for the movement of trains to less than two-thirds of that required ten years ago.

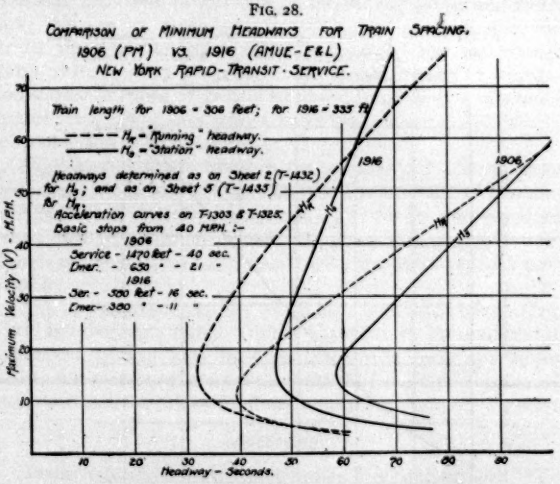
FIG. 27.



It is seen from Figs. 24 and 25 that as the station spacing is increased the traffic acceleration is also, which means, in turn, that the traffic volume for any given period is enlarged. This is for the reason that the time spent running at the maximum speed becomes a larger portion of the total time a train is on the road, and the average speed over a given district becomes correspondingly higher, as the station spacing is increased. The practice of having trains stop at alternate stations or groups of stations—"skip" stops, in other words—is based on this relation.

These curves also show that as the maximum running speed is increased the traffic acceleration becomes greater, but at a decreasing rate, until finally, at some critical speed value, it actually starts to fall off with a continued increase in speed. This is best illustrated by the curves for one-half-mile station spacing. The critical speed for the 1906 equipment and this station spacing is about 25 miles per hour, and that for the modern equipment about 35

FIG. 28.



The minimum headway required for train operation to-day is but two-thirds of that required ten years ago, due to the introduction of modern control equipment.
"Station" headway (H_S) determined as on Fig. 18.
"Running" headway (H_R) determined as on Fig. 21.
Figs. 26 and 27 are the bases for acceleration values.

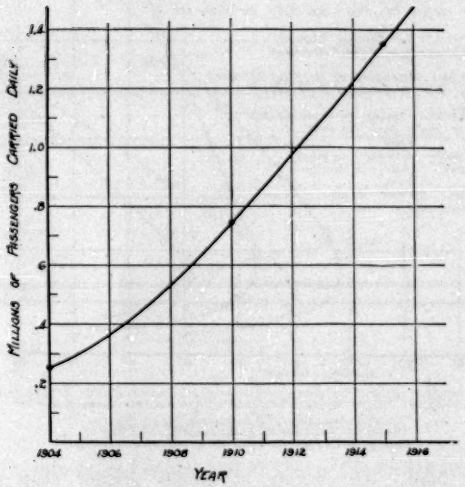
miles per hour. To operate at speeds greater than these critical values is actually to reduce the traffic acceleration and correspondingly the traffic capacity. As the station stops are spaced farther apart this critical value for speed becomes greatly increased. The reason for this peculiar relation is that the traffic acceleration ($\frac{V_s}{H_s}$), though directly proportional to the average speed V_s which increases with an increase in the maximum speed, is inversely proportional to the headway, H_s. A point

is finally reached where the increase in headway, H_s, due to higher speed is proportionately greater than the increase in average velocity, V_s, and the net effect on the quotient is to lower it. That is to say, the gain in average speed due to increased maximum velocity is more than offset by the increased headway, actually causing a lessened rate of traffic acceleration.

The relative performance curves on Fig. 24 show that modern

FIG. 29.

INCREASE IN NUMBER OF PASSENGERS CARRIED DAIL, - BY THE INTERBOROUGH RAPID TRANSIT COMPANY SINCE OPENING IN 1904.



brake apparatus as compared with that of 1906, has permitted an increase of 50 per cent. in *the number of trains* to be handled with the same roadway facilities in the way of number of tracks. While it is true that the modern motor equipment provides a higher maximum speed and a rate of acceleration somewhat better (14 miles per hour for the first 10 seconds, as compared with 10 miles per hour—see Figs. 26 and 27) the train length is also greater, con-

tributing toward an increased headway for the modern condition, other things remaining equal. This increased train length means, however, more passengers per train; therefore, the best overall comparison is found in the relative passenger-mile performance curves of Fig. 25. These show a gain in capacity of from 300 to 350 per cent., due to the use of modern equipment, with unchanged roadway facilities. This great advance in the transportation service, and, therefore, in the value of the railway properties which carry this service, depends on, more than on anything else, the strides made in the science of train control. For the present status of this science has rendered possible high rates of retardation started in trains of greatly increased length with a minimum loss of time, heavy cars with a large load capacity, and greatly reduced headway between trains.

Figs. 24 and 25 are based on the so-called old AMUE brake equipment, which involves an application reflex time of two seconds more than the latest equipment. The curves were prepared before this recent improvement was made and are, therefore, already not quite up to date in portraying the traffic capacity now possible.

Fig. 29 pictures the strides which have been made by the New York Interborough Rapid Transit Company in the number of passengers handled daily since its inception in 1904, and serves well as an interesting practical summary of the vital relation of train control to the value of railway properties.

CONCLUSION.

Trains cannot be moved unless they can be controlled. To turn loose the tremendous power of modern locomotives without adequate means for controlling it would be similar to generating a high steam pressure without suitable provision for containing it. The effectiveness of control will determine the speed and number of trains. It will determine the number of cars which may be successfully operated in each train, as well as their weight and variation in weight from the empty to the loaded condition. The word *determine* has been used. But, as a matter of fact, the advancement in the science of train control, arising in the types of apparatus designed to meet ever-changing requirements, has kept pace with, and is actually abreast of, the increasingly severe operating conditions. Therefore, the advancement in railroad effi-

ciency, which in this respect has placed us ahead of every other nation on the face of the globe, has been *permitted*, rather than determined, by the progress made in train control equipment.

Further advancement will also be permitted in just the same degree that the railroads continue to avail themselves of the advantages offered by modern equipment for train control; that is, amazing strides in furthering the economic worth of railway properties, to the public and stockholders alike, can be made by applying intelligently to the science of train control for solution of the weighty traffic problems of this day.

The attempt of this paper has been to point out more the *potential* value of, rather than the absolute necessity for, improved types of train control apparatus. This is not intended to be a claim that increasing the capacity of a railroad by any means will increase the supply of business, whether the means be double-tracking, improved methods of train control, or any other. Increased capacity for a railroad no more increases the business of the railroad than does the enlarging of a tank increase the water it originally contained. Putting in a second pipe line does not add to the content of an oil well—it merely makes possible the transportation of a larger output if the larger output be there. Similarly, providing improved traffic facilities for a railroad will not supply the traffic, but such provision will meet increased traffic demands. And all may rest assured that the demands will always precede the extension in capacity. In short, the science of train control solves *weighty* traffic problems. Where there is no problem obviously there is no solution required.

Unfortunately, the significance of the problems themselves, quite apart from their solution, is not appreciated by many who are directly concerned with them. There are evil results attendant upon the use of the single-shoe-per-wheel type of foundation brake gear; the use of air brake devices in service far beyond their designed capacity; the operation of trains with effective braking ratios widely varying from one portion of the train to another, due to leakage, lack of uniform piston travel, car loading, etc.—upon all of these and many other malpractices.

The extent of the evil results herein arising is beyond the ken of only too many whose interest it is to know of this indirect tax—this improperly invisible drain on the economic wealth of our transportation systems and, therefore, finally, on the commonwealth.

In other instances these results are accepted by many railroad managers as a matter of course; as necessary evils indissolubly associated with the operation of trains, and to be paid for as unavoidable elements of the cost of transportation.

The purpose of this paper has been to review the factors involved in railroad capacity and to show that train control can be made the most effective and profitable of all (in fact, it is so now); still, there is as much to be gained in this direction as has already been done. Some say we are getting along all right with what we have, and this may be granted, particularly as regards safe operation, but safety is now being had largely at the expense of economy and capacity. Is this wise? Is the investor satisfied? I think not when I see such efforts as are being made to increase capacity by "bigger power," greater capacity cars, etc.—the larger factor being neglected for the smaller.

What we now use generally is good, but what only a few are using is better, as they have proved. All the factors should advance at an equal pace if a rounded-out return is to be had. It is not the intent to condemn the old train control systems any more than progress in any direction may be considered condemnatory of that which has served its time and has been the pioneer of a better thing.

Wherever increase in capacity of a road is the desideratum it will pay to give the train control factor the most intelligent consideration. In other words, I intend only to set this up as a business proposition to be considered according to strict business principles.

It is high time that due study and thereby due appreciation be given to the underlying and, it is true, intricately interrelated causes for operating troubles, because their removal establishes an economic gain of a dual nature: First, the elimination of expense directly due to these troubles, such as damaged lading and equipment, delays, etc.; and, second, the extension in traffic capacity permitted without a corresponding increase in operating expense.

The adoption of adequate train control equipment will do more than any other means possible to remove operating troubles, and, with existing right-of-way facilities, to provide for the extension in traffic capacity which will make possible the realization of the utmost efficiency in that most wonderful of our industries, transportation by rail.

How a Large Manufacturing Company Disposes of Its Old Metal. J. M. BATEMAN. (*Proceedings of the American Institute of Metals,* September 11–15, 1916.)—One unaccustomed to the handling of raw material in large quantities little realizes the amount of work entailed in trying to dispose economically of the immense amount of waste incident to the carrying on of manufacture in a factory so large as the Hawthorn Works of the Western Electric Company. When, however, a monthly raw material input approximating $1,500,000 in value is considered it is not difficult to conceive the amount of waste resulting from its use. A somewhat clearer idea of the amount of material involved may be obtained from the figures representing the approximate monthly averages of the large items of junk disposed of. Brass, 100 tons; copper, 180 tons (125 tons from the cable, rubber and insulating shops); German silver, 17½ tons; iron and steel 89 tons; solder skimmings, 4 tons; lead, 833 tons. All these materials are carefully separated, cleaned, and baled in the form in which they command the highest price.

Experiment With Mercury Jet Interrupters. C. E. S. PHILLIPS. (*Proceedings of the Physical Society of London,* vol. xxvii, No. 4, August 15, 1916.)—The experiments here recorded were undertaken in the attempt to ascertain the form of the mercury column issuing from a hole in the side of a rotating drum that is continuously supplied with mercury by centrifugal action. Incidentally a new form of interrupter is introduced in which the interior is visible through a window in the lid. The arrangement forms a suitable apparatus with various forms of orifices and metallic contact segments, and an account is given of work in that direction. As it is important to insure the cleanliness of the mercury in interrupters of this type, the copper usually employed for the segments is replaced by tantalum which for many reasons is preferable. It is not "wetted" by mercury, it remains clean and bright indefinitely, and its high melting point renders it lasting.

Experiments with various forms of orifices are described, and it is pointed out that the issuing stream is only slightly affected by this means. An explanation is given of the fact that a vertical slit orifice will not produce a ribbon of mercury, and no matter how much the diameter of the orifice is increased beyond about 2 mm., the cross section of the mercury column remains unaltered. A method is described, however, by which a much larger stream of mercury can be obtained from the rotating drum if necessary. Brief reference is made to experiments with various volatile liquids in suppressing the flare which occurs when the mercury columns leave the contact segments. The three well-known substances which are most effective in this respect are coal gas, hydrogen, and ether. It is pointed out that experiments of this nature are necessary in view of the wide use now made of mercury interrupters in X-ray work.

ADMINISTRATION: ITS PRINCIPLES AND THEIR APPLICATION.*

BY

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INTRODUCTION.

1. OCCASION seems to exist for an attempt to clarify and systematize our ideas in this direction and to reach some acceptable conclusion as to the general nature of the fixed guides by which executive responsibility may most surely be discharged with success. We are living in an age which is making unprecedented demand for administrative ability—a demand not met by the supply of natively endowed administrators. There are, of course, a large number of such—both men and women—and there are, on the other hand, a large number of individuals who by reason of temperament, character, and education,[1] or what not, could never adequately fill an executive position, no matter how fully tutored to assume its responsibilities. But between these two extremes there is a much greater number of men and women than in either of the other classifications which, it is believed on the evidence of countless instances, represents latent capacity and is susceptible of easy development from the potential to the utilizable kinetic. It is this latter group which is especially addressed, although the born administrator will not suffer by having his ability defined and placed in intelligible and convenient order. The number of those who should be vitally interested may be closely approximated when we consider that the range of the applicability of administrative principles reaches from the household to national government itself, and that within a large administrative unit there are lesser units to be guided—first, in the interest of the institution's efficiency, and then as

* Communicated by the author and based on a paper read before the Summer Conference at the U. S. Naval War College, Newport, R. I., in 1913.

[1] NOTE.—The word "education," as here used, is not to be interpreted as mere book knowledge, but is intended rather to convey the idea of that breadth of view and understanding and mental grasp which is not denied the unlettered.

preparation for the assumption by the individual of higher and higher administrative positions—for promotion, in short. Such is the genius of all democratic institutions that no one is denied the rewards of demonstrated ability to handle men and wring results from an organization.

In the quest for information on the general subject of administration for purposes of study preliminary to its discussion it was a great disappointment to find that little was available. There were plenty of misleading titles in the indexes consulted, but the books and treatises under various captions involving the term "Administration" usually discussed "Organization," although in most there were here and there what may be characterized "Administrative gems." There is evidently considerable confusion as to the precise meaning and applicability of these terms, or, at least, an unsatisfactory differentiation between them. I am not sure that, in the following discussion, I will not be guilty of the same confusion in my view of which I am disposed to accuse others; but, if that proves to be the case, it may be explainable as concerning those points on the border-line of the subject where they blend almost imperceptibly into equally border-line points of organization. In any event, the writer's efforts will be gratifyingly rewarded if his readers are stimulated to think along the lines indicated, to test the truth and applicability of the enunciated principles, and, in the case of the proved administrator, to judge whether or not his success is in large measure attributable to an unconscious application of the enunciated principles.

As intimated above, the question of "Administration" is inextricably intertwined with that of "Organization," for nothing that is not organized can be administered, and nothing can be permanently organized that is not susceptible of administration. Yet the terms are not interchangeable and should not be confused, as they seem generally to be, if many articles on these subjects are any criterion. In spite of this interrelation, they are terms distinguishing distinct things and should be so separated in discussion.

Nothing conduces to this end so much as to define one's premises, and it seems well, therefore, at the outset to express my understanding of the key-words as a basis for what is to follow

DEFINITIONS OF KEY-WORDS.

2. The term "Administration," as comprehended in the subject of this paper and by which discussion will be guided, may be defined as: "The act (or acts) of administering; direction; management; government of public affairs; the conduct of any office or employment" (Century Dictionary).

While the term "Administration" is defined as above, narrowly, it always carries with it the idea of accomplishing its ends through the mediation of agents of various degrees and numbers, according to the kind and size of the organization, in behalf of the central figure—the integrating factor—the coordinating unit, who in turn represents the public directly.

"The administration of government, in its largest sense, comprehends all the operations of the body politic, whether legislative, executive, or judiciary; but in its most usual, and perhaps in its most precise, signification it is limited to executive details and falls peculiarly within the province of the executive department" (Hamilton, *Federalist*, No. 72).

"Sometimes (however) the term 'Executive,' which strictly means an authority which puts the laws in force, is opposed to the term 'Administration,' which (as above stated) implies the performance of every other sort of immediate government act . . . . . " (S. Amos, "Science of Politics," p. 99). It is this latter meaning which is accepted for present purposes.

3. "Organization" is the act of arrangement or *an* arrangement of related interdependent enterprises into a group with such system as will enable, and make it possible to compel, each to cooperate harmoniously with the others in the accomplishment of a common end.

There are two processes by which organization is attained—organization by accretion, *i.e.*, from without, as the growing of separate things into one; organization by differentiation, *i.e.*, from within, as exemplified in the biological development of the human race from a single cell. In the creation, upbuilding, and maintenance of most institutions both of these processes are operative.

Here we arrive at differentiation of the above terms by way of recapitulation. Organization is the substance; administration the form. Organization is the instrument; administration the method of employing it—the integrating, guiding, and energizing force.

4. Coming now to the term "Principle," I find it defined in the Century Dictionary as: "That which is professed or accepted as a law of action or a rule of conduct; one of the fundamental doctrines or tenets of a system . . . ." Expressing it in another way, "Principle is that on which something else depends; and this both for an original law and for an original element" (Sir W. Hamilton, Reid, Note A, E. 5, Supplementing Dissertation), and thus we see that there are both regulative and constitutive principles. It is only by very careful observation that we are able to extricate precisely the general law which is the expression of the regulative principle, and yet it is the given task not only to do this but to apply it.

DERIVATION AND HISTORY OF DEVELOPMENT.

5. In the beginning, of which the pioneer period of the United States may be taken as a near example, each individual worked in his own interest and was all-sufficient unto himself for the necessities of life. There was no relation or interdependence between individuals except as the family came into existence and supplied the source from which the administrative idea was derived, the male parent assuming authority.

Then came the stage in development which was marked by an accentuation of ability in one or another of the personal enterprises supplying the needs of life, and this change was accompanied by barter. *Pari passu* with the extent to which it was possible to make such personal arrangements, attention to the barterable article subsided on the side of custom (patronage) in the interest of giving more time and energy to the production of a barterable commodity or the development of a barterable ability of its own. And so with the increase of population and the more complicated demands of existence specialization in the social body progressed until each art or undertaking was represented by so many that competition supervened.

Then came the union of forces to supply the growing demand —a banding together of units for complete production in themselves without interdependence other than in certain obvious economies to be mutually realized, such as common workshop and business representative.

Following this came the division of the task among the members of the group wherein each unit of work constituted a part of

the whole only, and thus the idea of specialization was renewed, the size of groups increased, organization became imperative, and administrative systems were required. But the underlying philosophy in the old system of specialization was *laisser faire*. Each individual was left with the responsibility for doing his particular job practically as he thought best, with little or no help from the management. To-day a movement is on foot to bring the workman—the individual—out of this isolation and teach him to perform his part not only in such a way that it will dovetail with that of others but in accordance with the rules and laws of a science or art—Scientific Management.

6. By way of recapitulation we may construct a verbal triptych:

First, the individual, standing clear, sufficient unto all his simple needs, but beginning to develop a special ability.

Second, the individual lost in a maze of associated effort to meet communal requirements.

Third, the individual regaining his identity in training his abilities to the point of highest efficiency, while at the same time individual effort is being welded—not merely in associated, but interdependent and coöperative work.

Now that the last of these pictures is on the screen, we will at all times in the extension of civilization have in our midst examples of each one of these several successive steps in development.

7. The impelling influences to changed conditions in performing the world's work and meeting the growing demands and economic problems were increased efficiency and more rigid economy—the minimizing of waste, which were only to be had by the amalgamation of similar but interdependent industries or enterprises into corporate bodies under one management, calculated to secure harmonious coöperation in the accomplishment of a given end.

8. The principles of organization and administration have not been newly discovered; they have been rediscovered as in the history of so many unwritten fundamental laws governing the various relations in life. They were realized and given heed in the early Christian era and probably long before. Since then they have again and again exercised their influence and slipped

into disuse beneath the surface of the bubbling caldron, only to reappear and force themselves upon the struggling world.

In only one type of organization have they held uninterrupted sway—the military—and from this civil institutions have frequently derived their inspiration when formulating systems of administration.

The growth and effect of administrative machinery are attributed, therefore, to the enduring truth of

THE EVOLVED BROAD PRINCIPLES OF ADMINISTRATION.

9. "What deep joy fills the mind of the philosopher when, throughout apparently inextricable confusion, he can trace some great principle that governs all events, and that they all show forth" (Channing, "Perfect Life," p. 109).

Successful administration is not found in "the great personal or individual achievements of any one man standing alone and without the help of those around him." It is found in that type of coöperation in which each member of an organization "performs the function for which he is best suited (preserving his individuality and supreme in his particular function) . . . and yet is controlled by and must work harmoniously with other men" (Taylor). Such harmony of action, however, is not a happening of chance or without basis. It is assignable to a certain philosophy which has been wrought out of old knowledge so collected, analyzed, grouped, and classified that it constitutes a science. It is due to a mechanism, which has gradually evolved and each element of which is characterized a principle. Let us separate and examine the principles.

THE PRIMARY OR CONSTITUTIVE PRINCIPLES.

First.—A single mind in control from which the plan of action and the directing authority must emanate.

Second.—Subdivision of delegated authority in conformity with the branches of the organization.

Third.—The recognition of areas of discretion corresponding to subdivisions of authority, but within the range of loyalty to the end in view (mission).

Fourth.—The determination of a mission and the promulgation of a general scheme or plan for its attainment.

Fifth.—The exercise of a system of inspection.

These are formulated in deference to the motives which study and experience show to influence men. It is true that laws formulated on such a basis owing to the fact that the very complex organism—the human being—is concerned, "are subject to a larger number of exceptions than is the case with laws relating to material things." And yet the above principles are sufficiently broad to give play to flexibility. Moreover, "laws of this kind which apply to a large majority of men unquestionably exist, and when clearly defined are of great value as a guide in dealing with men."

10. *A Single Controlling Authority and Subdivision of Delegated Authority.*—The purpose of organization and administration as shown under the history of the development of the idea was increased efficiency and greater economy, and this could only be attained by concentrating under the leadership and harmonizing force of a directing centralized will. The provision for this office is the predominating feature of all modern organization, and the administrative principle it represents is given place throughout the lower reaches of every organization. Each division and subdivision has its head, and descent of authority and ascent of appeal pass through the subordinate heads of each step in the organization from or to the highest executive authority in the institution, although the head of each division or branch exercises authority over the subordinates in his branch of the establishment to whom he entrusts the execution in detail of the separate task, in the general plans, assigned him. He also assumes responsibility for discipline and results within the field of his supervision, exacting obedience on the one hand and guiding the activity of his immediate subordinates in such wise as will contribute most surely to the realization of the object decided upon in accordance with the plan formulated and transmitted to his subordinates by the executive chief, which in turn is dictated by the policy adopted by the management in its deliberations.

11. The object underlying this principle of administration is coördination, and its mainstay is discipline. Having subjugated the spirit of subordinates to the influence of the guiding intelligence, it remains to impose a common mission and advise as to the plan for reaching the objective. In fulfilling the conditions of this latter requirement where many divisions and subdivisions of work are to be coördinated it may be necessary to go into con-

siderable detail concerning the various tasks to be accomplished, even to the last link in the chain of administrative consequence. But in the distribution of tasks of progressively decreasing importance a double precaution must be exercised that each subordinate in the system of organization is not charged with greater responsibility than he can efficiently supervise—nor yet a task smaller than his capabilities. The one error of judgment defeats efficiency; the other defeats economy.

When professional capabilities and professional character are at fault in a subordinate, or if for any cause inadequacy to a reasonable standard of productivity develops and results fall short of expectations or needs, there are but two ways out of the situation, depending upon the nature and degree of the shortcomings: not by the meddling of a superior in duties which belong to another, but by instruction or by elimination of the inadequate individual and the substitution of one more capable—instruction where removal is impossible or there is hope of developing efficiency, " but for downright incapacity " wilful refractoriness and conscious disloyalty " there is only one remedy, and that is removal from office." In this connection, where it is merely a question of incompetency, the conception of a new division in the organization of an institution which is at once an employment office and clearing-house for personnel is based on the reasonable assumption that relatively few individuals are utterly useless; that every one has a particular niche in which he will work to the best advantage of himself and his employer; and that often an individual is wrongly placed on the occasion of his initial employment. If an employee is inadequate in his first assignment it may be found that he will give superior service in a second or third position, and it is worth the trial in the interest of justice to the individual (that he be not condemned off-hand) and in the interest of industrial economy. The idea is a new one, but experience with it justifies the belief that it is a practical instrument in the handling of men and is entitled to a permanent place in the industrial world. The " Scientific Employment Plan " was very interestingly presented some months ago by Burton J. Hendrick.

12. *Area of Discretion.*—As seen, the scheme of organization provides for lesser leaders—heads of divisions and subdivisions. It is here that the opportunity for so-called areas of discretion is

found, and it is necessary, in the interest of efficiency and economy, that advantage be taken of that opportunity to give play to initiative, which can be a distinct asset. The theory of the necessary existence of areas of discretion lies in the fact that unforeseen conditions and problems incident and collateral to the business in hand are continually arising in each branch of the work and must be met and solved without avoidable recourse to the higher administrative head. It also lies in the fact that the methods to be adopted in accomplishing a given part of a common task should ordinarily be left as far as possible in the hands of the division head. There is sound reason for this on the side both of the superior and of the subordinate.

In the case of superiors, while they may reserve the right to directly control as many details as they can efficiently handle, any undue interference in the field of responsibility assigned to subordinates absorbs their energies in illegitimate channels at the expense of their proper duties and thus jeopardizes the effectiveness of the general system. Such action upon the part of superiors constitutes confiscation of prerogatives and misappropriation of time and abilities and energy.

In the case of subordinates the consciousness that they are being held responsible for results alone and that their hands are free to direct the activities of their respective branches according to their judgment and ingenuity engenders a healthful sense of their importance to the big end in view and inspires an interest and loyalty which are valuable assets of the institution.

In the one case the central administrative hand is infinitely strengthened by confining himself to the conceded limits of his office. In the other case repeated interference from above dampens ardor and reduces service to pure perfunctoriness, if nothing worse.

It is, of course, sometimes, in emergency, necessary to assume temporary control outside of the area of superior action which the administrator may have outlined for himself, but occasion for this should arise only in an emergency during the absence of the subordinate concerned or when the skill and reliability of a subordinate are in question.

Similar to the limitations of central administrative activity, the area of discretion of subordinates is also limited. It is limited to "the right to do *right* as you (the subordinate may) please"—

the right to act and exercise judgment within the range of loyalty to the policy and mission and plans of the institution with which he is identified.

13. *A Common Object and General Scheme.*—Here, in turn, and finally, we come to the thing for which organization and administration are brought into being and perfected—a common purpose or a common object; the something of mutual concern in the interest of which coöperation is demanded. Herein is found the force that binds superior and subordinate. The underlying basis of liberty of action in subordinate positions is "loyalty to the scheme, and this, of course, demands that there be a scheme to be loyal to," and that the scheme be intelligibly communicated through all the ramifications of the organization.

14. *Inspection.*—Inspection is a procedure upon which, as a stimulant, the continuous effective and guiding activity of the principles of administration in great part depend, and yet, as it is an important part of every administrative system, it will be discussed at this time and may itself be accounted a principle, as so much of success rests upon its proper exercise. It is one of the sources for the gratification of the "honor-motive" in *esprit de corps,* to be discussed later; it stimulates a persistent positive activity and defeats that otherwise progressive passive lapse from established standards of efficiency and economy; it maintains the objective of the administration clearly before the members of the organization; it constitutes a channel to publicity; it furnishes comparisons which inspire the manager to seek to improve methods and enable him to alter his standards of maximum and minimum performances, both for the individual and for the several branches of the organization; and it brings to bear on the administration a vision and understanding not dulled by the familiarity incident to daily contact with the subject or matter, but acute to defects and praiseworthy features alike and capable of discerning opportunity for improvement.

15. The agencies of inspection are not limited to a formal office, and none should be neglected. "It is not enough for the head of a system of administration to express his resolution in regard to a proposed action" (Rodgers, p. 17). In addition, the execution must be supervised to insure active fidelity to the plan, and in small institutions the principal himself will be able to fill the office unaided. He must, however, always be a party (albeit

an independent party) to the function of inspection, and in this fact is emphasized the prerequisite that the administrator should be familiar with the technical details of his organization.

16. The condition of the relations between interdependent internal branches of the organization and between those and the interested public; the periodic conference of heads of divisions, together with the chief, and their daily reports to him concerning unusual or important matters or occurrences; the statistical comparative study of other similar administrations; and the use of publicity are other forms of inspection automatic in their assistance to the administration. But, valuable as those are in their limited scope, none of the foregoing contributions to the principle of inspection relieve the urgency of the demand in large institutions for the exercise of the function by an inspection office or staff which shall be independent of the strictly executive branch and beyond the control of its subordinate members, and through which the administrator may keep in touch with every branch of his responsibility.

CONDITIONS UPON WHICH THE PRINCIPLES OF ADMINISTRATION DEPEND. THE SECONDARY OR SUPPORTING PRINCIPLES.

17. "The mechanism of management must not be mistaken for its essence or underlying philosophy. Precisely the same mechanism will in one case produce disastrous results and in another the most beneficent. The same mechanism which will produce the finest results when made to serve the underlying principles of scientific management will lead to failure and disaster if accompanied by the wrong spirit in those who are using it" (Taylor, p. 128). Though this truism was thus expressed by one of the foremost exponents of scientific management, it is in its very neglect to give adequate practical recognition of the fact that scientific management has failed. Many of those using it have displayed too single-minded an interest in material ends—have regarded men as mere machines, and have too often overlooked the human factor involved, which, "as in everything, is essential." Applied psychology is a closed book to them and to most administrators. "Unhappily, it is our habit not to study the psychology of the peoples with whom we have to deal, but to watch and react upon the shifts of circumstances" (E. J. Dillon, *The Fortnightly Review*, November, 1915).

The Spirit of the Administrator.—Laws, rules, and orders are important and unavoidable, but they should be limited to such as may be absolutely necessary without tying the hands of the governed too closely, and human nature must be taken into account in their formulation. Too many rules produce confusion and invite disobedience, and, moreover, rules do not of themselves accomplish good administration. The character of the administrator determines that. The personal element comes in here and is reflected in the administration. The administration is the gauge of discipline, and the degree of discipline is the expression of the quality of the personal element. The wish and tendency of the normal individual are to do right, and there is little fickleness about such a material thing as an organization. When affairs go wrong, therefore, it is very much more apt to be the administration, not the organization, which needs improvement or correction.

18. There are certain underlying conditions which emanate from the spirit of the man (administrator) upon which successful administration depends. This is shown not only in his bearing toward his subordinates and the manner in which he approaches the functions of discipline and inspection, but the consistency which he observes in his personal conduct and administrative control of others.

19. *Forms of Ceremony.*—The recipient of authority in any system of administration must not be left unaided in that authority. Reference is not here made to rules and regulations, which should only be employed as a last resort, but to moral force, which is expressed in the widely observed influence of one man over another, based on a conception of greater ability in one by the other—an unconscious intellectual assent to superiority, and which is preserved by certain forms of ceremony tending to exalt the prestige of the executive head—some, the outward show, well recognized; the nature of others less appreciated.

20. The former consists of *artificial aids*—the stringent prescription of forms of respect in military life, such as differences in uniform, the salute, and the guard of honor, and in civil life the etiquette and courtesies demanded of one (the effect of which is sometimes enhanced by little difficulties of access) in his intercourse with an official superior. These are not vain show. They are customs, the observance of which is obedience in spirit, and "they conduce to discipline as conventional good manners . . ."

21. The other forms of ceremony which are conditions to the continued operation of the dormant administrative principles involve: *qualities of temperament*, such as patience and self-control; *qualities of mind*, such as consistency, self-reliance, and a lively sense of justice; *habits of conduct*, such as dignity and reserve of bearing, though not stripped of "the amenities of cordiality or the cultivation of good cheer" within proper limits; and the practice of keeping one's own counsel to the extent at least that one's deliberations, doubts, and official acts may not fall "unnecessarily under the scrutiny of one's subordinates." It is given to few men to decide a question of moment correctly in off-hand fashion without some deliberation. Patience and decision are two very important qualifications of an administrator, but they must be carefully adjusted, for they are in a sense antithetic, and the former, if exaggerated, will produce the effect of weakness. There should be no appearance of seeking approval or of dependence upon the views of subordinates, although encouraging them to express their opinions. The aim should be, rather, to lead them to seek advice and to regard the administrator's invitation to share his counsel as a privilege; to develop that mutual confidence—the feeling that they are all working for the same end and will share in the results—which should exist between a leader and his men.

Every administrator of large institutions who does not come directly in contact with a majority proportion of the personnel, particularly the lower ranks, must be careful in what manner he indicates his disposition of kindliness toward them. It is to be presumed that he will at least deal with them along the line of plain justice. His personal and official acts at the beginning should demonstrate his intentions, and his consistency in that direction should be a matter of course. There should be no demagogic appeal for popularity by an offer to adjust all difficulties. The assurance is but "a flash in the pan" in the direction of promoting contentment. The attempted execution of the promise breeds unrest and ultimately discontent—the very opposite to what was intended. But this is not all. The offer is translatable into an invitation to insubordination, which, in the first accounting, brings trouble and difficulties in the path of those upon whom he depends for the efficiency of the subdivisions of the organization and, in the last accounting, brings trouble and

ignominy upon himself—if not because of the failure of his administration, then because he is brought face to face with the impossibility of redeeming his promise—because, sooner or later, he is discovered as unable to stand with his head straight and his hands clean by the test of his own estimate of a "square deal." In one grand stroke such a man announces himself a panacea for all ills—and little ills at that—which increases their number, when he might to more purpose leave some of these little ills for the subordinate leaders to straighten out and fare infinitely better in so doing. "You can fool some of the people all of the time and all of the people some of the time, but you can't fool all of the people all of the time." To announce such a purpose as that of which I have been speaking may be justified in politics, but it is hardly far-seeing honesty or conducive to continued smooth running in an administration.

22. These ceremonial elements and ceremonial "don'ts" indispensable to successful control of an organization represent the spirit and are inseparable from the practical form of the function. They may seem at variance with the prevalent *spirit of democracy*, but there is that within the human nature of everybody, no matter what his station in life, which is favorably excited by a certain degree of pomp and show, and when this is governed by tact and discrimination (without bombast) it may be exercised effectively within the bounds of toleration and well short of resentment by even the strongly socialistic. The differentiation of self need not be obtrusive, and, while preventing that fatal suspicion among subordinates that the leader is no better than they, it "secures an ascendency which would not otherwise be procurable."

23. *Discipline.*—The impersonal attitude of mind toward offenders in the exercise of disciplinary authority is extremely important—not only as being the proper attitude, but as a means to easing discipline and robbing punitive measures of their undesirable sting. Most men who dance are willing to pay the fiddler when their attention is called to the debt, but in inflicting punishment it should be done with the idea of teaching a wholesome lesson and in such a way as to deny the victim any possibility of raising the thought that authority is arbitrary and tinged with animus or of engendering a lingering sense of resentment and an idea of persecution. Past offences cannot be corrected; future offences can be prevented. Punishments should be awarded

solely to minimize the recurrence or commission of the same or similar offences by the same or other individuals. "Between the flourish of the 'Big Stick' and the appeal to 'Sweet Reasonableness' lie ages of progress. At just what point the use of force yields to the plea of justice, as the more effective means of guiding human conduct, it may be impossible to determine. But that there is such a point in social evolution, and that we have long since passed it, must be evident to those who will carefully weigh the evidence of history" (*The Public*, vol. xvii, No. 825, p. 73).

24. *Regulations and Rules.*—The administrator's relation to the established order, and rules and regulations necessary in any management of affairs, exhibits the rationale of the impersonal attitude of mind. He is not the government or the management (the board of governors or directors in a corporation). He is but the representative thereof—in such rules as he may promulgate himself by virtue of authority vested in him, as well as in those placed in his hands as from higher authority. In the first case his rules are in behalf of the management as certainly as if prepared by the management itself. Such rules, as soon as established, become the rules of the management, whether specifically approved by it or not, and any infraction of them constitutes an offence against the government or management and not an offence against the person of the administrator. It is true that some offences are so flagrant that the emotion of anger is difficult to control, but anger has no place in the official relation of an administrator and his subordinates, and it had better be given opportunity to subside before punishment is adjudged if justice is the aim. There is as much difference between forbidding certain acts and stimulating participation in right conduct as there is between a God who punishes sin and one who rewards virtue. In other words, "It is easier to maintain law and order where men look upon the law as a friendly guide than where they see in it only brute force."

25. *The Spirit of Inspection.*—Among many individuals the duty of inspection seems to inspire in them a new and strange spirit—new in the sense that it is abnormal to the degree of being antagonistic to their native feelings for fellow-officials or coworkers; strange in that they seem to be burdened with the necessity of unearthing some culpable error or omission, instead of

looking for evidences of efficiency and taking note of the features in which there seem opportunity for, or the possibility of, improvement. They seem to become possessed of the idea that the fulfilment of their task—the mark of their efficiency in the duty imposed—is unfavorable criticism. They make their office an unpleasant instead of a grateful duty—a thankless instead of a helpful duty—a source of irritation instead of an encouragement. It all depends upon the attitude of mind in which the duty is undertaken and the tasks approached—the interpretation of the purpose of the office, whether or not it is a successful inspection and an element in fostering administrative cohesion and loyalty. In the one case it puts the head of an administration and all under him (in the field to be inspected) on the defensive and resistant in the direction of the inspector's inquiry. In the other case it brings to the inspector's assistance the coöperation of the inspected and an openness and frankness in all matters of concern to the inspector. Such a reception would be accorded because of the hope of deriving the benefit of an eye fresh to the task of ferreting out defects overlooked by the too familiar eye of those who are constantly on the ground and, therefore, without perspective—because of the hope of eliciting approval or suggestions for improvement. It is the larger view which should be taken by the inspector, and he will then find himself in the best strategic position to carry out all the requirements—even the disagreeable, if so it be—of his office.

26. *Consistency in Policy and Executive Acts.*—There can be no question of the importance of consistency—in policy and executive acts—to successful administration. Efficiency to the extent that it rests upon contentment and sound progress cannot be secured without consistency, for that idea reflects certainty of aim and brings mutual understanding, which are at the bottom of coöperation. It makes little difference what a given policy may involve: if consistently followed, progress and development are inevitable. It makes little difference how rigid the discipline—how taut the requirements: if the discipline is consistent it will be a happy institution. Everything is run to-day as it was yesterday and will be to-morrow, and all hands know what to expect. At all times everybody knows what can and what cannot be done—what is expected of him and what somebody else will do—and can adapt himself accordingly.

27. *Esprit de Corps.*—Thus far the physical and intellectual and moral elements, the latter, so far as the principle is concerned, constituting and supporting administration, have been briefly discussed. There remains to be considered the collective moral element represented by the personalities of the subordinate members of the organization—the agency by which the administrative system is vitalized. The moral element exerts a most important influence for weal or woe, but usually the former under normal conditions and when the natural spirit of men is not perverted by extraneous interference. It is *esprit de corps*, or, in psychology, a force known as "motor-activity"—"the self-organization of groups"—which lies at the bottom of this collective moral element. It is the primitive disposition commonly manifested among men in association—"the blest tie that binds"—and has everywhere developed a code of "unwritten law and traditions more potent than statutes and regulations in welding into a consistent whole the somewhat incongruous elements" that go to form any organization like a large industrial institution, or an army, or a navy, or a nation, for that matter. "It *means* a revival of the principle of coöperation as against the principle of competition"—of unity of effort as against "the disrupting tendencies of opposing or selfish interests"—and is exhibited in the movement which is tending naturally towards integration, federation, and scientific management.

It does not embrace trade unionism which is essentially artificial—nor is it in any sense altruistic. Dr. Colin Scott argues, as abstracted by Ward,[2] that "the everyday experience of life shows that the efficiency of a man as a social animal or as a member of the community depends less upon his individual attainments than upon his power of adapting himself to his social environment." Within well-recognized bounds the cult of individualism is important as a stimulus to the fullest development of the best possibilities of one's native characteristics and capabilities, but it loses its value and means worse than nothing if it goes too far and does not in the end make for a more intelligent and harmonious collectivism—if it becomes translated into license and fails in deference to that essential community force which tends to hold every one in some relation to others and makes one's

[2] Ward, "The Psychological Bases of Esprit de Corps," *United Service Magazine*, N. S., 40, p. 390.

every act of greater or less negative or positive consequence to one's associates.

28. *Esprit de corps* is *engendered* by two motives, subscribing to the above observation—the desire to perform efficient service to one's group, and the desire to receive appreciation. If, as has always been recognized, *esprit de corps* is an enormous power in the interest of efficiency, binding associated individuals closely to each other and to the organization, and if the desire to do and the "honor-motive" are the two great motor-activities leading to *esprit de corps*, then those two motives should be planted and stimulated in the microcosm of every organization as valuable aids to their effective administration.

29. But when developed through the agency of these motives how is *esprit de corps* to be *preserved?* Even in this subtle commodity the principle of reciprocity—the give-and-take of life—is operative. The benefit of *esprit de corps* is an unconscious contribution to the welfare of the organization at the hands of the organization's collective membership, yet the "sympathy, enthusiasm, devotion, and jealous regard for the honor of the body as a whole" which it implies must be paid for. This is best done with the strengthening sustenance (also subtle) of just and kindly, though in no sense philanthropic or patronizing, attention by the administrator to the spiritual and material interests of the individual members, lest *esprit de corps* become transmuted into the trade-union spirit. It must be nourished by a gratification of the underlying motives to its origin. It must be nourished by such a manifest personal concern for the health, contentment, and progress of the men as will give ungrudging response to reasonable requirements in surroundings and conditions—as will offer a helping hand, should occasion require, and lead the individual to feel that he is an important part of the whole and a partner in its success.

And it must be nourished by tangible benefits appearing to emanate, and actually emanating, directly from the increased efficiency. The matter of profit-sharing is not as popular as it should be from this point of view—nor, so far as is known, has it ever been properly applied or theoretically developed to its feasible perfection. It involves loss-sharing as an inseparable counterpart. The operation of one should imply the operation of the other, but through the influence of artificial forces economic benefit and penalty have not been equally distributed. Profit has

been largely on the side of capital; loss largely on the side of labor. This is the genesis of labor's organization for protection and a preventable danger to *esprit de corps*.

30. *Publicity*, as another condition upon which the continued activity of the principles of administration depend, ranges itself with *esprit de corps*. *Esprit de corps* is not devoid of the possibilities or, even, likelihood of fault, and it may operate against efficiency unless carefully watched. It is the dangerous disposition it embodies, as a reflection of the perfectly proper administrative protection of individuals, to view loyalty in too personal and narrow a manner and to shield the unworthy, which must be guarded against. The influence of publicity serves this purpose. Publicity of one sort or another helps to place the object of our loyalty in relief and give it perspective, so that false *esprit de corps* becomes discernible and the threatened injury to the cause, in which all must share, is made clear.

THE ART OF APPLYING ADMINISTRATIVE PRINCIPLES.

31. The fundamental principles of administration, although derived from military institutions, "are applicable to all kinds of human activities, from our simplest individual acts to the work of our great corporations, which call for the most elaborate cooperation"—even to all social activities. The theory of their application is based upon the idea of unquestioned authority on the one hand and willing subordination on the other, with hearty coöperation throughout. This theory is just beginning to be understood, while the function itself has been a gradual evolution over a long period of years. However, be this as it may, no administrator can afford to overlook *the social democratic tendency* of the times, which, grafted on the long-existing spirit consequent upon "the doctrine of the political equality of all men," makes his task more delicate. He must be sensible of the real existence of this tendency as something to be reckoned with and adjust his methods accordingly. It is in this indirect way and to this extent only that cognizance can be given to the prevailing idea of social democracy in administrative operations, for its principles cannot be harmonized with the paramount principles of administration—the dominance of one directing power, with its counter-requirement, obedience. Autocracy in administration alone insures permanent efficiency—autocracy tinctured with an enduring sense of its many-sided obligation to others—not in-

toxicated with a consciousness of its power, and any refractory or unsympathetic spirit should be cut out summarily in its incipiency —not as a punitive act, denying the right of opinion to others, but as a preventive measure—a measure calculated to preserve harmony and insure coöperation—a measure in the interests of the unimpeded execution of the task. The idea of democracy may come into being and be properly conceded a place outside and above the administrative head—namely, in the authority which outlines the policy to be preserved and the objective toward which the whole organization directs its energies.

32. "Only the man who himself knows how to obey, who has learnt from personal experience how grievous an inopportune or superfluous order can be, and how inexpressibly hard it is, in such a case, to resist the impulse to revolt—only such a man will avoid blunders when he is himself in a position of command" (Von Spohr).[3] The art of command is to elicit a cheery and willing obedience; not to compel a slavishly servile obedience. "It is the first alone which conduces to happiness . . . , insures a firm, unshaken discipline, and inspires men." So act as to rob obedience to an order of any semblance of servile submission, and this is best done by indoctrination. Orders must not be thinkingly or unwittingly used as a means to magnify one's own importance, and when necessary they should be unobjectionable in both matter and manner. Moreover, no more orders than are absolutely essential should be issued, for unless heed to this injunction is given all independence, all initiative, and all love of responsibility, so valuable in subordinates, will be killed.

33. Like the disposition to feel that nothing can go on without an order, any fondness for domineering must be checked, as this "leads to tyranny and incites insubordination." "What is wrong must be reproved, but not severely, not sharply, not in the form of censure, and an equal readiness to praise must be shown." "No man likes to be severely found fault with (particularly in the presence of others), but everybody is willing to accept instruction and does better another time. The man who has cause to fear fault-finding forswears initiative . . . and plays safe or keeps in the background."

34. With regard to order writing or written instruction, "every superior that finds he has been misunderstood should first

[3] Von Spohr, "The Art of Command," *Journal of the Royal Service Institute*, vol. 53, 1909, p. 56.

look for the fault in himself . . . ." The superior may know well enough what he wishes to order, but the question is: Do his words adequately express the idea—does his order or instructions—written or verbal—convey his intent clearly and concisely? No one can edit one's own expressions even in moments of quiet and leisure, and much less so at times of stress and excitement.

THE SUBORDINATES IN AN ADMINISTRATIVE FIELD.

35. Those who have carried a dinner-pail and worked for day's wages and also been employers of labor will tell you rightly that there is something to be said on both sides of this question of administration. The direction of obligation is by no means solely from above downwards. The obligation of the subordinate is just as pressing, and recognition of it will spell personal success in almost direct proportion to its contribution to institutional efficiency—the obligation to be loyal to a trust, to act promptly, to concentrate his energy, to do the things given him to do no less well than he may. This paper cannot properly be closed without an attempt to urge upon the subordinate a sense of his responsibility in the game. He must be a dependable unit if he is to continue to play his part, but it is important to all concerned—himself included—that he be something more than just dependable. No one can honestly separate his own success and advancement from the success and advancement of his employer, but it is easy and proper for an institution determined upon progress to separate itself from a slipshod, inattentive, indifferent, and half-hearted employee. Learn initiative, develop a capacity for independent action, and show a willingness and readiness to put your shoulder to the wheel for the benefit of all. Efficiency is not attained by accident. It is the offspring of imagination, interest, intelligence, industry, discipline, enthusiasm, and determination. The three greatest of these are the first two and the last, if we assume intelligence. The others will follow. As for discipline, no individual can afford to shy at it and no institution, however democratic, can neglect it. Neither the individual nor the group can succeed without it. The word is popularly and erroneously supposed to mean the sacrifice of private judgment—of a conceived right to do as one pleases. What it really means is a full-grown will-power (one of man's greatest, but most neglected, possessions), self-control, and a fitness to live and work

among one's kind, and it constitutes the basis of that team-work which accomplishes big results, in that it makes it possible for each individual to contribute his full quota to the general prosperity. There should be no inertia in a subordinate requiring the prompting of his chief. This immediate superior should be convinced by action that assigned duties or tasks will be performed efficiently and expeditiously; that he may look with confidence for results; and that, once an order is given, he may dismiss the matter from his mind in the interest of other constantly appearing, pressing problems, in the knowledge that the matters so intrusted will in due time be placed before him in completed form.

The administrative head of a large establishment is too busy to have to bear in mind or keep before him the hundred-and-one matters under his jurisdiction connected with its operation. He has a right to expect his subordinates to exhibit a sense of responsibility, and to relieve him of such a necessity, once he has given an order or expressed a desire and explained his purpose in either. Remember that the subordinate's area of discretion is his right to do right as he pleases—his right to pursue his own methods so long as he is loyal to the aim of his superior and gives the results desired—" delivers the goods," according to specification and in expeditious fashion.

In conclusion it may be said that one's bearing in relation to others and the attitude one assumes toward a given task will go far to determine success or failure. A determination to do the best one can will open up unexpected vistas of interest and inject an enthusiasm into a seemingly dull and prosaic field of duty.

" The very fact that one is exercising his own ingenuity, searching for new avenues of usefulness, and giving some service from his powers to originate beyond the strict letter of the . . . (requirements), gives birth to a subtly gratifying sense of contribution to that part of the public weal in which one's lot is cast; to a feeling of partnership—even proprietorship—in the increasing efficiency of the . . . (establishment)."

" If to do were as easy as to know what were good to do, chapels had been churches, and poor men's cottages princes' palaces." These are words which Shakespeare put into Portia's speech. But I am of opinion with Thomas C. Galbreath that " to know what ' were good to do ' is not easy; it is far more difficult than doing it after one knows," and I have, therefore, been encouraged to present my observations.

NOTES FROM THE U. S. BUREAU OF STANDARDS.*

REPORT OF THE BUREAU OF STANDARDS IN THE CASE OF ECONOMY FUSE AND MFG. CO. VS. UNDERWRITERS' LABORATORIES, INC., CONCERNING THE FIRE AND ACCIDENT HAZARD OF THE ECONOMY REFILLABLE FUSE.[1]

[ABSTRACT]

THIS report represents the results of the investigation carried out by the Bureau of Standards acting as referee on the joint request of the Economy Fuse and Mfg. Co. and Underwriters' Laboratories, Inc., on the question of the relative fire and accident hazard of Economy Refillable fuses and fuses at present listed as standard by Underwriters' Laboratories, Inc.

The joint appeal was made to the Bureau under date of May 17, 1915, and the question submitted for decision was as follows:

"Has it been shown that the use of the fuses manufactured by the Economy Fuse and Mfg. Co. results in no greater fire or accident hazard than the use of other cartridge enclosed fuses at present listed as standard by Underwriters' Laboratories, Inc.?"

The evidence on which the finding of the Bureau was based includes a large number of tests of fuses under widely different conditions as well as inspections of numerous fuse installations in practice, personal interviews with many fuse users, evidence and arguments submitted by the Economy Fuse and Mfg. Co. and Underwriters' Laboratories both at a public hearing and by correspondence, and evidence and arguments submitted by a number of manufacturers of fuses at present listed as standard by Underwriters' Laboratories.

During the course of the Bureau's investigations it was shown that the Economy fuse when new and properly filled or refilled operates satisfactorily under the most common working conditions of overload and moderate short circuit, and it was also shown that this fuse possesses some marked advantages over the approved fuses with which it was compared. The Economy fuse

* Communicated by the Director.

[1] Technologic Paper No. 74.

is, however, distinctly inferior to most of the approved fuses under severe short circuit conditions. It has not yet been established that it will not introduce hazards peculiar to refillable fuses owing to deterioration from repeated blowing of the fuse elements in the same casing and possibly from long-continued subjection of the fuse to the working current. The approval of the present type of Economy fuse for unrestricted use would therefore result in a lowering of the standard of fuse performance under severe test conditions and might introduce hazards in actual use the importance of which it is difficult to estimate at this time.

The experience with the present type of Economy fuse is not yet sufficient to determine whether the total hazard is greater or less than it is with approved fuses as they are actually used in practice. The investigation, therefore, leads to the following finding:

"It has not been shown that the use of the fuses manufactured by the Economy Fuse and Mfg. Co. will result in no greater fire or accident hazard than the use of enclosed cartridge fuses at present listed as standard by the Underwriters' Laboratories, Inc.

"On the other hand, the evidence in the case does not show that the use of Economy fuses has on the whole resulted in any greater fire or accident hazard than is involved in the use of standard enclosed cartridge fuses.

"In comparison with fuses listed as standard by Underwriters' Laboratories, the fuses at present manufactured by the Economy Fuse and Mfg. Co. have been shown to possess certain features which tend to increase the hazards involved in the use of fuses, and other features which tend to reduce such hazard. The relative importance of these features can be determined only by extended experience under working conditions.

"It is therefore recommended that Economy fuses be not approved at present for general use on the same basis as fuses at present listed as standard by Underwriters' Laboratories, Inc., but that a continuation and extension of their use be permitted by municipal and underwriters' inspection departments under conditions where their performance can be observed by each inspection department until sufficient experience regarding their performance under service conditions can be obtained to justify an unqualified approval or refusal to approve."

The evidence on which the finding has been reached is summarized and discussed in the report. The report contains about 350 printed pages including numerous tables and 110 oscillographic records, showing the performance of both Economy and approved fuses under various short circuit conditions.

THE CONSTITUTION AND MICROSTRUCTURE OF PORCELAIN.*

By A. A. Klein.

[ABSTRACT]

A PETROGRAPHIC microscopical study of porcelains prepared in the laboratory of the Bureau of Standards and of commercial porcelain, as well as of various combinations of the raw materials which enter into porcelain, has led to results which are interesting and important both scientifically and technically.

Bodies and mixtures of the following types were examined: kaolin, feldspar-kaolin, feldspar-quartz and feldspar-clay-quartz. These were burned to various known temperatures. The commercial bodies investigated represented the practices of the following countries: United States, England, Germany, France, Austria, Denmark and Japan. The end in view was to obtain data concerning the change involved by burning porcelain at various temperatures, and it was found possible to correlate to a certain degree the constitution and microstructure with the burning temperature of bodies whose composition lay within the limits of whiteware and hard fired porcelains.

The result of this investigation leads to the following conclusions: Kaolin appears homogeneous microscopically when heated up to 1200°. At about this temperature a trace of dissociation occurs. As the temperature is raised above 1200° the dissociation increases very slowly at first, then at an increasing rate, until at 1400° it seems to be complete. The products of dissociation are silica and aluminum silicate. The latter compound has been identified as an amorphous phase of sillimanite from the following facts: It shows no crystalline form, has an index of refraction above 1.60, and by heating at a higher temperature (about 1450°) it inverts to minute needle crystallites corresponding to sillimanite in all determinable optical properties.

* Technologic Paper No. 80.

Up to 1340°, in mixtures of quartz and feldspar, the quartz dissolves to only a small extent in the feldspar glass. At 1460° the quartz is practically completely dissolved in specimens having as high a quartz content as 50 per cent. quartz to 50 per cent. feldspar.

At 1340° in specimens containing kaolin and feldspar, the kaolin dissociates entirely. The amount of crystallized and amorphous sillimanite increases with an increased content of kaolin at least to a concentration of 50 per cent. kaolin to 50 per cent. feldspar.

At 1460° apparently 10 per cent. kaolin is entirely soluble in the feldspar glass. With higher concentrations of kaolin the amount of crystallized sillimanite increases. The needle crystals are well developed and comparatively large.

At 1310° in quartz-clay feldspar bodies, the feldspar is present as a glass; the clay shows almost complete dissociation with the formation of amorphous sillimanite mainly and but little crystallized sillimanite, while the quartz is undissolved and the grains may still be of considerable size, up to .2 mm. or more, depending upon the fineness of grinding.

By burning these bodies at 1380°–1400°, the feldspar glass dissolves considerable quartz, there being only a comparatively small amount of quartz remaining. The quartz grains are much rounded and etched, and they seldom show a length over .06 mm. The clay is dissociated with the formation of crystallized sillimanite, although an extremely small amount of amorphous sillimanite may be present.

The changes involved by burning commercial bodies are identical with those of laboratory-prepared bodies. Commercial ware ranges from a low-burned porous whiteware in which, except for the dehydration of the clay, only the feldspar is changed, to very high-fired porcelain which consists of glass, sillimanite crystallites and more or less of residual quartz. The quartz grains observed in the whiteware and in the low-fired vitreous ware are large and angular, showing a size of .2 mm. or more, whereas in the hard porcelains, due to solution, the quartz grains are rounded and etched, and seldom exceed .05 mm. in length.

The constitution and the microstructure of porcelain depend upon the temperature of burning, and change as this temperature changes. This has served as a basis for the estimation of the

probable burning temperatures of the commercial bodies, which was accomplished with success, the error involved being within 25 degrees. It appears that the time of burning factor is by no means as important as that of the burning temperature in determining the constitution and microstructure of the ware.

No cristobalite or tridymite has been definitely observed in any of the laboratory or commercial bodies examined. It appears that the quartz dissolves in the feldspar glass more readily than it inverts to the other modifications of silica.

In conclusion, it may be stated that the petrographic microscopic study of porcelain led to interesting and, it is to be hoped, important technical results. It has placed the chemical and physical processes involved in the formation of porcelain on a more quantitative thermal basis. Furthermore, it has offered a means of estimating the burning temperature of a ware from the examination of a fragment much too small in size to be satisfactory for even a chemical analysis.

THE FAILURE OF BRASS.—2. THE EFFECT OF CORROSION ON THE DUCTILITY AND STRENGTH OF BRASS.*

By Paul D. Merica.

[ABSTRACT]

RESULTS of investigation made on a homogeneous, alpha brass, have shown that the electrolytic solution potential of this material is increased by the application of a tensile stress. This increase, as measured, amounts to approximately 0.1 millivolts for 10,000 lbs./sq. in. of stress.

Using this fact as a basis, an explanation can be given of the decrease of strength and ductility of brasses when corroded while under stress. Over a roughened surface of a bar under tensile stress, this stress will vary in value, being greatest at the bottom of furrows and depressions and least, almost zero indeed, at the tops of the ridges. The EMF will therefore, other things being equal, be greater; *i. e.*, more electropositive, at the bottom of these furrows than elsewhere; corrosion will set in here most rapidly forming a crack, which will grow narrower and sharper, its rate of growth being greater, the sharper it is. In time the cross-

*Technologic Paper No. 83. Complete copies of the paper may be obtained by application to the Bureau of Standards, Washington, D. C.

section of such a bar is so reduced by these cracks, that fracture occurs, the brass failing apparently at a stress value less than the ultimate strength, and exhibiting only light elongation (ductility).

This explanation is borne out by the examination of a number of brass failures, which have occurred under such conditions.

THE FAILURE OF BRASS.—3. INITIAL STRESS PRODUCED BY THE "BURNING-IN" OF MANGANESE BRONZE.*

By Paul D. Merica and C. P. Karr.

[ABSTRACT]

IN connection with the failure by cracking of a number of manganese bronze valve castings in the Catskill Aqueduct, at or near areas repaired by "burning-in," an investigation has been made of the initial stress produced in a manganese bronze double bar casting, by the burning-in of a constrained portion. The stresses measured were in each case about 8000 to 10,000 lbs. / sq. in.; *i.e.*, the true elastic limit of the material, and the material within the burned-in area was of course in tension. The microstructure of the portion, adjacent to the burned-in metal was not altered; the burned-in metal was in all cases of finer grain than that of the casting.

The conclusion is reached that, although distortion of a burned-in casting may partially relieve the initial stresses set up by this operation, such castings will in all probability, generally, contain local stresses of dangerous magnitude; *i.e.*, near the elastic limit of the material. Castings, repaired in this manner, should then, either be thoroughly preheated or subsequently annealed in order to eliminate these stresses, which are responsible for later failure by cracking, as described above.

THE DETERMINATION OF THE DEGREE OF UNIFORMITY OF BARS FOR MAGNETIC STANDARDS.*

By Raymond L. Sanford.

[ABSTRACT]

MAGNETIC standard bars are used for the calibration of permeameters and the comparison of methods of magnetic testing with a standard method. One requisite of a magnetic standard

* Technologic Paper No. 84. A copy of this paper may be obtained by application to the Bureau of Standards, Washington, D. C.

* Scientific Paper No. 295.

bar is that it shall be magnetically uniform along its length. If this condition is not met, errors may arise which cannot be calculated or eliminated from the measurements, and which may be of considerable magnitude.

The degree of magnetic uniformity of a bar may be determined from observations of the distribution of magnetic leakage along the length of a specimen when it is magnetized between the poles of a suitable electromagnet. The degree of uniformity is indicated by values of the rate of change of leakage along the length of the bar. Deviations of these values from a constant indicate the presence of non-uniformities. An increase in the value indicates a magnetically hard spot while a decrease indicates a soft spot. The degree of uniformity can be indicated by means of a curve plotted between values of the rate of change of leakage and displacement along the length of the bar. This curve is called the uniformity curve and for a uniform bar is a straight line parallel to the axis of displacement.

Errors in magnetic measurements due to non-uniformities depend upon the induction, the nature, location, magnitude and extent of the non-uniformities and upon the type of apparatus. For a given accuracy, no ordinate of the uniformity curve must deviate from the mean by more than a given amount.

This method may be applied to the examination of magnetic materials for mechanical inhomogeneities and the detection of flaws.

THERMOELECTRIC MEASUREMENT OF THE CRITICAL RANGES OF PURE IRON.*

By George K. Burgess and H. Scott.

[ABSTRACT]

THE methods hitherto employed for the determination of the thermoelectric properties of conducting materials possess the characteristic, which is particularly disadvantageous in the case of a substance such as iron, which has two critical ranges, in requiring a length of the material in question to have a temperature distribution from the maximum to the lowest. There may then be ambiguity or superposition of thermoelectric effects.

* Scientific Paper No. 296.

Using a length of pure iron wire (Fe = 99.968) of some 7 cm. length and 0.5 cm. diameter joined between the hot junctions of two Le Chatelier thermocouples within a furnace 60 cm. long, several series of accurate observations in vacuo of the thermoelectric power of the couple Iron-Platinum have been taken, at 2° intervals, over the temperature range 0 to 1000° C.

The critical point A_3 is marked by a discontinuity of considerable magnitude in the thermoelectric power vs. temperature curve at about 915° C. on heating and 900° C. on cooling. At A_2 there is a change in shape of the curve. The thermal effect at A_2 is superimposed upon the thermoelectric and manifests itself as a slight protuberance or dent at 768°.

The following table gives the thermoelectric power (dE/dt) Peltier effect (TdE/dt) and Thomson effect (d^2E/dt^2) for iron-platinum from 0 to 1000° C.:

Thermoelectric Power of Iron Against Platinum.

Temperature Centigrade t	Microvolts per degree dE/dt	Peltier Effect TdE/dt	Thomson Effect d^2E/dt^2
0	19.5	5320	−0.010
100	18.1	6750	−.027
200	15.4	7280	−.035
300	11.7	6700	−.033
400	9.5	6390	−.010
500	9.1	7030	+.009
600	10.8	9430	+.026
700	14.3	13910	+.036
780	18.1	18980	+.045
800	18.4	19740	+.014
880	19.4	22350	+.010

	Heating	Cooling	Heating	Cooling	Heating	Cooling
900	19.7	17.5	23100	20510	.000	−.400
910	19.4	10.8	22940	12770	−.050	−.040
920	16.6	10.9	19800	13000	−.575	+.010
930	11.4	11.1	13710	13350	−.023	+.017
1000	12.6		16030		+.017	

These thermoelectric observations give further evidence of the distinct character of the critical points A_2 and A_3 delimiting the regions of alpha, beta, and gamma iron.

LIQUID MEASURING PUMPS.*

By F. J. Schlink.

[ABSTRACT]

The importance of the measuring pump is principally due to the large quantities of gasoline sold as fuel for motor cars, by far the greater part of this commodity being sold through the various forms of measuring pumps. About 27 firms are engaged in the manufacture of the various measuring pumps and liquid dispensing systems.

The principal advantages of this type of apparatus lie in its convenience in delivering gasoline directly to the tank of the automobile, with a minimum of evaporation, spillage and fire hazard.

The first and most common type of measuring pump is the piston pump, which is in principle the usual plunger pump with stops which act to define accurately the limits of the piston stroke, and consequently the volume generated by the piston on each cycle. Both single-acting and double-acting piston pumps are used for measuring purposes. In addition to the pumps with reciprocating pistons, pumps with rotating pistons are used, the application of these at the present time being more familiar in the so-called molasses pumps, which are especially adapted to the handling of viscous liquids.

Most reciprocating and rotary piston pumps are provided with a counter or tally to indicate the number of strokes or cycles which have been completed. Such counters should be so designed and located that they will operate correctly, indicate unmistakably, be actuated only very near the conclusion of the stroke or cycle, and be plainly visible to and readable by both merchant and customer.

In a second type of apparatus, the measurement is performed by passing the liquid through a meter, commonly of the nutating-piston type, the discharge of liquid through the meter being produced either by displacement of the oil over water, by pneumatic pressure, or by mechanical pumping. With the type of meter now used, it is important that the rate of flow through the meter be maintained as nearly constant as possible, and the liquid stream must be kept free from solid particles and sediment, or the meter regularly and carefully cleaned.

* Technologic Paper No. 81. Read at the Eleventh Annual Conference on Weights and Measures of the United States, May 23, 1916.

A third type of measuring system is founded upon the principle of overfilling a measuring chamber, supplying liquid until an excess is present over the nominal delivery, and then removing the excess by abstraction to a definite level. The liquid may be supplied to the measuring chamber by air pressure, by evacuating the chamber with a vacuum pump, or by direct mechanical pumping from the supply tank. The abstraction of the excess may be performed either by gravity through an overflow pipe, or by siphoning, the height of the liquid remaining being determined by the vertical height of the face of the orifice of such pipes above the bottom of the measuring chamber. In a modification of this type, the measurement is obtained by direct reading of the height of the liquid column on a graduated scale, at the beginning and end of the withdrawal of the liquid from a tank or cylinder. The cross-sectional area of such tanks must be limited in order that the volume value of the scale interval be not excessive.

A portable apparatus is also in use, comprising a small storage tank and a measuring pump, the whole arrangement being adapted for moving to the place of filling most convenient to the purchaser. Such devices favor a higher precision than most stationary tanks in that the suction lift can be kept at a minimum, and the danger of vaporization and leakage below the piston is much reduced.

In the testing and inspection of measuring pumps it is essential to determine whether the installation is free from inward leaks of air and outward leaks of gasoline. Leaks of either kind are fatal to accurate measurement, and any adjustment made before the correction of defects of this kind cannot be expected to be permanently effective. Much trouble of this nature can be prevented by following with scrupulous exactness the instructions of the manufacturers, which are provided to guide the owner in making the installation.

The principal causes of short delivery of pumps of the piston type are leaking of foot valves, and formation of vapor or air space under the piston, resulting from excessive virtual suction lift or leaky piping, or piping containing traps tending toward the periodical retention of vapor. It appears that a total vertical suction lift greater than 7 feet may be excessive with the ordinary commercial (not "blended" or "casing-head") gasolines. "Casing-head" or "blended" gasoline should not be lifted any appreciable vertical distance on the suction side in those types of

pumps in which the piston and cylinder form the measuring element, on account of the vaporization difficulty already mentioned. Special installations, in which the pump is located at or near the level of the supply tank, should be employed for the last-mentioned grades of gasoline.

Another source of error which should be guarded against in design and installation of piston measuring pumps is the inertia of the moving liquid column.

The use of long filling-hoses, and of hoses so arranged that they cannot be readily drained, results in cases of short delivery to individual purchasers; this should be obviated by so relating the length of hose and the height of hose connections that the hose can always be readily drained to a definite level. A measuring pump should not be installed inside a building with the hose connection outside, unless so arranged that the manner of operation of the pump is clearly visible to the purchaser. The writer also questions the advisability of permitting the use of a shut-off cock on the outer or delivery end of the hose, as its use may readily operate to prevent the regular drainage of the hose into the customer's tanks.

It is recommended that the design of piston type measuring pumps should tend toward types which cannot be operated through less than full strokes without that incompleteness of the cycle being clearly indicated to the customer. This certainty of stroke completion is readily obtainable in simple designs, which so far as the writer knows have not yet been offered for use.

In order to prevent the short deliveries consequent upon operating the piston pumps at less than full stroke, they might well carry a clearly legible placard or notice, reading "This Pump to Deliver—Gal. for Each *Full* Stroke."

Pumps which provide for the return of undelivered liquid to the storage tank should be equipped with some manner of device which will prevent this return taking place at the same time that liquid is being delivered to the purchaser, as such diversion of a portion of the customer's purchase constitutes an easy means to fraud.

In testing pumps, the hose should be removed if this is feasible, and the test delivery made from the pipe outlet, unless a suitable can-filling outlet is provided. Test should be made with a standard measure of suitable capacity and of the conical type,

used in combination with a cylindrical graduate reading to cubic inches, or by a measure of special form, similar to that designed by Mr. Theo. Seraphin of Philadelphia, which comprises a zero graduation for the correct discharge, and additional graduations for deliveries in excess and efficiency, so arranged as to cover the normal range found in the field work. Two speeds at least, one faster and one slower than normal operation, should be used in the test in order to determine whether or not the action of the pump is substantially independent of the speed of operation.

It is recommended that piston measuring pumps which have been standing unused for a considerable length of time be operated for one or two strokes to eliminate the first stroke short delivery which often occurs on account of valve leakage, vaporization, thermal contraction, etc.

Proper sealing of the pump by the weights and measures official means not merely the attaching of a sticker or tag to the pump, but the actual sealing by wires and soft metal seals of all accessible portions of the pump upon which adjustments or alterations can be made which will affect the volume of the pump discharge. The manufacturers will be glad to furnish drawings and instructions relative to the points of adjustment on their own makes of pumps. The manufacturers should be required to provide all their pumps with suitable drilled holes or devices to facilitate this sealing by the weights and measures official.

A careful, full and explicit record of the test should be made. The complete paper contains a form well adapted for this purpose, which owing to lack of space cannot be included here.

Results of the tests in the field indicate that pumps in service show a decided tendency toward undermeasurement, this being due to the fact that nearly all of the defects of construction or installation tend to produce errors in this one direction. These include: leaks of air and liquid, retention of liquid by the hose, vapor formation consequent upon excessive suction head, failure to complete the full stroke, and slippage of liquid past valves and piston.

The Bureau will endeavor to maintain a full and up-to-date mailing list of manufacturers of measuring pumps, and will supply these names upon request. Particular inquiries on matters in this field, referred to the Bureau of Standards, will be given careful attention.

NOTES FROM NELA RESEARCH LABORATORY.*

ON PHOTOELECTRIC PHOTOMETRY.

By Jacob Kunz.

In the present field of photoelectric investigations four principal groups may be distinguished. The first group is concerned with the fundamental problem of the relation between the velocity of the electrons emitted and the frequency of light; the second with the relation between the intensity of light and the photoelectric current; the third with the normal and the selective effect; and the last with the influence of gases. The present investigation deals with the second problem. Several physicists have attacked this question with widely disagreeing results. Elster, Geitel and Richtmeyer found a linear relation between the two quantities; Griffith, Lenard and Ives on the other hand found non-linear relations; Ives especially found a large variety of curves. It was the purpose of the present investigation to find out the true relation and an explanation for the discrepancies among the previous results.

The photoelectric cells used were of different types. The largest number investigated were of spherical shape as described in a recent number of the *Physical Review*. The anode was a circle across which there were wound fine platinum wires in order to create an approximately uniform field. The larger part of the sphere was covered with the sensitized alkali metals, sodium, potassium, and rubidium; only a small hole of about 8 mm. diameter was left free for the entrance of the light. The intensity of light was varied by moving the source to and from the cell, or by means of Nicol prisms, or by means of a rotating sector disk. The current was measured by means of a sensitive galvanometer or by means of a large constant resistance and the electrometer, by the accumulated charge or by the drift method.

When the galvanometer was used the light intensities were rather high, and the result was without exception a curve concave toward the light intensity axis, but it was observed that the curve approached a straight line, when the intensity and the hole through which the beam of light entered the photoelectric cell

*Communicated by the Director.

was decreased. With an opening of about 5 mm. practically straight lines were obtained. The same result was also obtained with the other three independent methods. In the majority of cases the highest point was about one or two per cent. above the straight line, a point which will be further investigated. The light intensity varied from 1 to 36. The results are indicated by the following two series of observations. I = intensity of light, i = photoelectric current.

i	I	$\frac{I}{i}$	i	I	$\frac{I}{i}$
9.0	12.8	142	7.15	11.1	155
34.6	48.8	141	15.9	25	157
70.9	101.3	142	28.5	44.5	156
106.5	151.5	142	63.6	100	157
139.6	200	144	98.6	156.2	158
186.6	268	144	128.2	204	159
226.3	326	144	175.7	278	158
251.8	360	144.1	212	331	157
276.7	397	144.3	262.2	400	154

As resistances were used a cadmium sulphate solution in a capillary tube about $\frac{1}{1000}$ normal, a graphite resistance as used by Ives and a photoelectric resistance. The graphite resistance was not very reliable. According to its surface conditions one can obtain, with the same cell and with the same light intensities, curves of widely different character, concave, convex, and irregular. But it is occasionally possible to have the resistance in a very good condition and then straight curves are obtained with the doubtful exception of the highest point.

Other types of cells have been constructed which gave, however, less satisfactory results. In one cell an approximately uniform field was created between two plane parallel electrodes, the lower being made of alkali metal, the upper of silver wire netting. A third parallel electrode was at the same time introduced in order to apply a counter field. Ives himself has claimed to obtain, without giving evidence, a straight line by means of a large spherical field; a cell constructed according to his prescription, gives approximately a straight line, but not so distinctly as the small cells, described above. If light is applied to the cell without a potential difference, then the light interrupted and the field applied, there was found a considerable photoelectric current.

Nela Park, Cleveland, Ohio.
October 9, 1916.

THE EMISSIVE POWER OF TUNGSTEN FOR SHORT WAVE-LENGTHS.

By E. O. Hulbert.

IN this investigation the variations in the emissive power of tungsten have been determined between 3200 A° and 5500 A° for true temperatures, according to Worthing's scale,[1] from 1746° K to 2785° K. Preliminary spectrograms of a tungsten strip through a quartz window at various temperatures showed an appreciable amount of radiation as far as 2900 A°. The light from the tungsten in a glass bulb was studied quantitatively by means of a grating spectograph with a sodium photo-electric cell and electrometer. The relation between the energy incident on the cell and the deflection of the electrometer was found to be a linear one for monochromatic radiation. The transmission of the glass, hot and cold, was determined; the measurements were corrected for this effect and for the finite width of slits.

The change in emissive power with wave-length was determined by measurements on a black body at the melting point of paladium, 1822° K, and on the tungsten at a temperature 2143° K. Conditions were the same in the two sets of measurements except as regards to slit widths. Because of this difference and because of some scattered light which was later noticed in the spectra, the black-body calibration has been used only in determining the relative changes in the emissive power with wave-length. In the table the absolute values given depend on the value .470 for wave-length 4677 A° at 2143° K (from unpublished measurements by Worthing). The emissive power is seen to increase with decrease in wave-length.

TABLE I.

True Temperature 2143° K

$C_2 = 14460$ deg. $x\mu$

Wave-length	Emissive Power of Tungsten
3478 A°	.470
3717	.487
3956	.484
4196	.495
4435	.466
4677	.470
4916	.460
5158	.453
5400	.447
5641	.436

[1] A. G. Worthing, JOUR. OF FRANK. INST., **181**, p. 417, 1916. *Phys. Rev.*, **7**, p. 497, 1916.

The change in the emissive power with temperature was determined by drawing the "isochromes," *i. e.* the curves for a constant wave-length showing the relation between the logarithm of the energy emitted and the reciprocal of the temperature. The isochromes were found to be straight lines within the error of experiment. Calculations based on the determinations of the slopes of these lines showed that the emissive power increased as the temperature decreased for all the wave-lengths under investigation. The data further indicated that the rate of increase of emissive power with decrease of temperature became greater as the wave-length became shorter.

Nela Park, Cleveland, Ohio,
October 3, 1916.

Fixing a Photographic Plate Before Development. ANON. (*Revue Generale des Sciences*, vol. 27, Nos. 15–16, August 15–30, 1916.)—At first sight it may appear absurd to think of fixing a photographic plate before the latent image has been developed, because fixing consists in dissolving the reduced bromide of silver. When an exposed sensitive plate is treated with hyposulphite of soda the opaline coating becomes completely transparent; not the least trace of an image is discernible, and the gelatine appears to contain nothing that can be disclosed by photochemical action. Nevertheless, paradoxical as it appears, this method is not only theoretically possible but has also several practical applications. If it were not limited to plates greatly over-exposed (six to eight times normal exposure), it would solve, better than any other process, the problem of development in open light.

The plate, protected from actinic light, is first immersed in a 2 per cent. solution of hyposulphite of soda. This bath dissolves the bromide of silver much more slowly than the usual fixing bath of 20 to 25 per cent. strength, but the latter has the disadvantage of destroying the delicate half-tones. Dissolution requires about 30 to 40 minutes, according to the thickness and hardness of the gelatine. The plate can then be developed in full daylight. If desired, the plate may be washed and dried and laid aside for future development. The image is disclosed by *physical development*, that is to say, by bringing a reducing agent and a soluble salt of silver in contact with the exposed film. The developer will contain, for instance, pyrogallol and nitrate of silver. The pyrogallol decomposes the nitrate of silver, and the silver so precipitated is deposited upon those points of the sensitive coating which have been exposed to the light and in quantities proportional to the exposure. The exposed points constitute in fact so many centres of attraction for the deposit of silver which are progressively reinforced.

NOTES FROM THE RESEARCH LABORATORY, EASTMAN KODAK COMPANY.*

THE MEASUREMENT OF THE ABSOLUTE VISCOSITY OF VERY VISCOUS MEDIA.[1]

By S. E. Sheppard.

[ABSTRACT]

WHILE the fall of a denser body through a liquid is frequently used by industrial chemists as a measure of viscosity, the values obtained are usually taken as *fixed empirical standards*, neither absolute nor relative viscosity coefficients being derived. It is pointed out that Stokes' equation for falling bodies enables these to be determined from readily measured quantities, the equation being expressed in the form:

$$K = 2/9\, R^2 (S - S')\, g/U.$$

where K = coefficient of viscosity.
R = radius of sphere.
S = density of sphere.
S' = density of liquid.
U = terminal velocity of fall.

An important condition is the ratio of the diameter of the vessel to that of the sphere. This ratio, say γ, should be very large if the equation is to give correct results. In the paper of which this is an abstract details are given of an experimental investigation dealing particularly with this factor. Steel ball bearings of ½ in., ¼ in., and ⅛ in., true to ∓. 02% were used, falling in a very viscous nitrocellulose solution. It is found that if T be the time of fall over a fixed interval L in the steady state when $U_{\text{lim.}} = L/T$, when the values of T diminish steadily as the ratio γ increases, approaching a constant limit for very great values of γ. An empirical equation is obtained for the function $T = f(\gamma)$ of the form

$$T = T_K + C/(\gamma - 1)^2$$

where T_K is the true time of fall for $\gamma = \infty$ and C is a constant for the experimental conditions. The correspondence of the calcu-

*Communicated by the Director.

[1] Communication No. 40 from the Research Laboratory of the Eastman Kodak Company.

lated values with the experimental data is exhibited in curves. The expression is compared with one obtained from theoretical considerations by R. Ladenburg (Ann. Phys. 23, 287, 447, 1907) which gives somewhat lower values for the viscosity, but is only valid on one side of the range. The expressions for Stokes' law in the case of cylindrical vessels, corrected for the influence of the wall, are:

$$K = \frac{2/9\, R^2}{L} (S\text{-}S') \frac{T}{1 + 2.4/\gamma} \qquad \text{(Ladenburg)}$$

$$K = \frac{2/9\, R^2}{L} (S\text{-}S') \; T - C/(\gamma - 1)^2 \qquad \text{(Sheppard)}$$

THE ELECTROLYTIC DETERMINATION OF SILVER FROM CYANIDE SOLUTIONS.[1]

By A. S. McDaniel and L. Schneider.

[ABSTRACT]

THE electrolytic determination of silver from cyanide solutions as described in manuals on electro-analysis is subject to relatively large errors and therefore cannot be used in its present form for the assay of certain photographic raw materials and products where a good degree of accuracy is of prime importance. If it were not for these errors the method would be specially convenient for this class of materials.

The method has been carefully standardized and the different sources of error identified and their magnitude determined. The plus errors are due mainly to inclusions produced by colloids. The minus errors are due to cathodic corrosion and losses in washing. It has been shown that under certain conditions, especially with certain classes of material, the plus and minus errors almost exactly balance, while under other conditions either the plus or the minus errors predominate. These facts account for the widely varying results obtained by the method in the hands of different analysts. A procedure is given for overcoming these errors. This results in a highly satisfactory method both as to accuracy and speed. The accuracy obtainable suggests that with certain further refinements the atomic weight ratios of the halogens and of nitrogen to silver could be determined by the analysis instead of the synthesis of the appropriate silver salts.

[1] Communication No. 19 from the Research Laboratory of the Eastman Kodak Company.

THE FRANKLIN INSTITUTE

(Proceedings of the Stated Meeting held Wednesday, October 18, 1916.)

HALL OF THE FRANKLIN INSTITUTE,
PHILADELPHIA, October 18, 1916.

PRESIDENT DR. WALTON CLARK *in the Chair.*

Additions to membership since last report, 25.

The following communication from the Board of Managers was presented:

"The Board of Managers at its regular meeting held September 13, 1916, confirmed the action of the Committee on Stocks and Finance in disposing of the property 1911 N. 18th Street, representing a portion of the investment of the Bloomfield H. Moore Memorial Library fund, and requests the Institute to ratify and approve the action by passing the following resolution: 'At a regular meeting of The Franklin Institute held October 18, 1916, a quorum being present and voting, it was unanimously, *Resolved*, That the sale of premises 1911 North 18th Street, heretofore made to Gertrude Mieterer for the sum of $3000, be ratified and confirmed, and that the action of the President and Secretary in executing a deed for the said premises unto the said Gertrude Mieterer, be approved.'" The resolution was passed unanimously.

Mr. Charles E. Bonine, Chairman of the Committee on Science and the Arts, reported the condition of the work of the committee. The Chairman called attention to the appointment of a National Research Council by the National Academy of Sciences on the request of the President of the United States, and stated that the purpose of the Council was to bring about further co-operation between existing governmental, educational, industrial, and other research organizations, with the object of encouraging the investigation of natural phenomena and the increased use of scientific method in strengthening national defense. He requested the Institute to authorize the appointment by the President of the Institute of a special committee to aid the Council in any line of work it may conduct and in which it might desire the co-operation of the Institute, whereupon the following resolution was adopted:

Resolved, That the President of the Institute be authorized to appoint a Committee of the Institute to co-operate with a similar Committee of the National Academy of Sciences for the further organization and encouragement of scientific research in the United States.

Mr. Charles E. Bonine, Chairman of the Committee on Science and the Arts, then introduced Mr. Carl E. Akeley of New York City, to whom had been awarded the John Scott Legacy Medal and Premium by the City of Philadelphia, on the recommendation of The Franklin Institute, for his Cement Gun. This apparatus is used for applying cement mortar and stucco,

the material being applied in the form of a spray upon the object to be coated.

The President presented the medal and accompanying certificate to Mr. Akeley, who returned his thanks for the honor conferred upon him.

Mr. Bonine was again recognized, and introduced Mr. John Van Nostrand Dorr of New York City, to whom had been awarded the John Scott Legacy Medal and Premium by the City of Philadelphia, on the recommendation of The Franklin Institute, for his Hydro-Metallurgical Apparatus, namely, the Dorr Thickener, the Dorr Agitator, and the Dorr Classifier, which have lately been used extensively in the cyanide industry.

The President presented the medal and accompanying certificate to Mr. Dorr, who expressed his thanks for the recognition of his inventions.

After the transaction of the above business, a joint meeting was held with the Philadelphia Section of the American Institute of Electrical Engineers, Mr. H. P. Liversidge presiding.

Dr. Charles P. Steinmetz, Chief Consulting Engineer, General Electric Company, Schenectady, N. Y., addressed the meeting on "Scientific Research in Relation to the Industries." The speaker stated that as engineering is an applied science, the engineering industries naturally depend for their success upon the use of correct scientific principles and the application of newly discovered scientific facts. Since established research institutions were not able to furnish all the information desired for use in the industries, industrial establishments have recently gone into scientific research with vastly extended facilities, with the result that of late the largest number of scientific advances have been made in industrial research laboratories. In Germany it was first realized that the establishment of the industrial research laboratory was one of the most profitable investments which could be made by the large industries, but in the last few years American corporations have also fully awakened to a realization of this fact, and the research laboratories established in this country by industrial corporations are doing scientific work of the highest order in chemistry, physics and mechanics.

Numerous illustrations of research methods, and of the value of the results of industrial research were given.

After a brief discussion a unanimous vote of thanks was tendered the speaker and the meeting adjourned.

R. B. OWENS,
Secretary.

COMMITTEE ON SCIENCE AND THE ARTS.

(Abstract of Proceedings of the Stated Meeting held Wednesday, October 4, 1916.)

HALL OF THE FRANKLIN INSTITUTE,
PHILADELPHIA, October 4, 1916.

DR. GEORGE A. HOADLEY *in the Chair.*

The following reports were presented for first reading:

No. 2638.—Continuous Automatic Centrifugal.

No. 2674.—Rubber Hysteresis Tester. This latter report was made advisory and adopted.

The following reports were presented for final action:

No. 2675.—Sharples Super-centrifuge. Action deferred.

No. 2676.—Composite Low Expansion Electric Lamp Leading-in Wires. Elliott Cresson Medal to Dr. Byron E. Eldred, of New York, N. Y., adopted.

R. B. OWENS,
Secretary.

SECTIONS.

Mechanical and Engineering Section.—A meeting of the Section was held in the Hall of The Institute on Thursday, October 5, 1916, at 8 P.M.

Mr. Charles Day occupied the Chair.

W. J. Humphreys, C.E., Ph.D., Professor of Meteorological Physics, U. S. Weather Bureau, Washington, D. C., delivered a lecture entitled, "The Atmosphere and Aviation." Dr. Humphreys explained the relation of the circulation of the atmosphere to gravity, temperature contrast and earth rotation, and gave brief accounts of trade winds, monsoon winds, cyclones, anti-cyclones, etc.

A practical method was shown for estimating the approximate direction and velocity of the wind, which may be expected at flying levels under any condition of the weather.

Those phenomena of the atmosphere which the aviator knows as "air pockets" or "holes in the air" were also discussed.

Colonel George O. Squier, Ph.D., Aviation Section, U. S. Army, late Military Attache, American Embassy, London, England, War Department, Washington, D. C., delivered a lecture entitled, "Efficient Military Air Service." Colonel Squier gave a very brief outline of the tentative plans thus far developed for producing the necessary matériel and the training of the personnel of the Army Air Service.

After an interesting discussion by Mr. Frederick W. Barker, President of the Aeronautical Society of America, and other gentlemen, a vote of thanks was extended to Dr. Humphreys and Colonel Squier.

Adjourned.

T. R. PARRISH,
Acting Secretary.

Section of Physics and Chemistry.—A meeting of the Section was held in the Hall of The Institute on Thursday evening, October 12, at 8 o'clock, with Dr. George A. Hoadley in the Chair. The minutes of the previous meeting were read and approved.

Louis Vessot King, A.M., ScD., F.R.S.C., Associate Professor of Physics in McGill University, Montreal, Canada, delivered a lecture "On the Acoustic Efficiency of Fog-signal Machinery." The various forms of fog-signal machinery, and the researches conducted with them, were reviewed in historical order. A description was given of the extensive research conducted by Dr. King at Father Point, Quebec, in September, 1913, in which a Northey "diaphone" or siren served as the source of sound, and a Webster "phonometer" served for the measurement of the intensity of the sound at distances as great as 8 miles from its origin. The various conditions which influenced the efficiency of the diaphone were discussed. The lecture was

illustrated with lantern slides, and with the apparatus used in the research.

After a discussion of the paper, a vote of thanks was extended to Dr. King, and the meeting adjourned.

JOSEPH S. HEPBURN,
Secretary.

MEMBERSHIP NOTES.

ELECTIONS TO MEMBERSHIP.

(*Stated Meeting, Board of Managers, October 11, 1916.*)

RESIDENT.

Mr. Edward L. Clark, Engineer, Commercial Truck Company of America, 27th and Brown Sts., Philadelphia, Pa.

Dr. Thomas E. Eldridge, Physician, 1811 North Logan Square, Philadelphia, Pa.

Mr. Robert E. Glendinning, Banker, 8862 North 32nd St., Philadelphia, Pa.

Mr. C. Stuart Patterson, Banker, 1000 Walnut St., Philadelphia, Pa.

Dr. Max J. Stern, Physician, 711 North Franklin St., Philadelphia, Pa.

Mr. Robert L. Wood, Civil Engineer, Cruse-Kemper Co., Ambler, Pa.

NON-RESIDENT.

Mr. James G. Detwiler, Chemist, The Texas Company, Port Arthur, Texas.

Mr. Armin Elmendorf, Instructor in Mechanics, University of Wisconsin, and for mail, 311 North Murray St., Madison, Wis.

Mr. W. E. Fletcher, Chemical Director, Atlas Powder Company, Wilmington, Del.

Dr. Frederick W. Getman, Chemist, Hillside Laboratory, Stamford, Connecticut.

Mr. Zay Jeffries, Metallurgical Engineer, Department of Mining Engineering, Case School of Applied Science, Cleveland, Ohio.

Mr. Paul M. Kuehn, Assistant Engineer, The Bartlett Hayward Company, Baltimore, Md.

Mr. William J. McGurty, Chemist, The Bartlett Hayward Company, Baltimore, Md.

Mr. Benjamin Estill Shackelford, Physicist, Westinghouse Lamp Company, Bloomfield, N. J.

CHANGES OF ADDRESS.

Mr. Richard L. Binder, 931 North 8th St., Philadelphia, Pa.

Mr. Harold V. Coes, Hotel Westminster, Toronto, Ontario, Canada.

Mr. R. Burdette Dale, Chief Mechanical Engineer, Erie City Iron Works, and for mail, 1150 Brown's Ave., Erie, Pa.

Mr. E. L. Gibbs, 502 King St., Jacksonville, Fla.

Mr. William J. Hammer, Cor. Palmer and Ellison Aves., Bronxville, N. Y.

Mr. Tinius Olsen, Aldine Hotel, 19th and Chestnut Sts., Philadelphia, Pa.

Prof. I. M. Rapp, 536 Boulevard, Norman, Okla.

Mr. Ernest Stutz, 922 Madison Ave., New York, N. Y.
Mr. Clarke Thomson, Merion Station, Merion, Pa.
Mr. William Chattin Wetherill, 307 West Springfield Ave., Chestnut Hill, Philadelphia, Pa.
Mr. Otto C. Wolf, N. E. cor. 31st and Jefferson Sts., Philadelphia, Pa.
Mr. Howard Sellers Worrell, 36 E. Montgomery Ave., Ardmore, Pa.

NECROLOGY.

Mr. Mathias Pfatischer was born in Bavaria on September 4, 1858, and died at Roselle, N. J., September 10, 1916. He came to the United States in 1882, and for twenty-seven years was Consulting Engineer for the Electro-Dynamic Company, at Bayonne, N. J.

Mr. Pfatischer devised an electric stearing gear for battleships, and designed and installed the electrical equipment on the steamships St. Louis and St. Paul, including the ship's telegraph and other novel features. Among his most important inventions is the Inter-pole Variable-Speed Motor, for which he was awarded the John Scott Medal by the City of Philadelphia in 1908, on the recommendation of The Franklin Institute.

Mr. Pfatischer was a member of the leading electrical societies of this country. He became a member of The Franklin Institute in 1889.

Mr. Thomas E. Baird, Villa Nova, Pa.

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Mr. Charles A. Rutter, 411 Walnut St., Philadelphia, Pa.

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CURRENT TOPICS.

Origin of the Open-hearth Process. H. H. CAMPBELL. (*The Iron Age*, vol. 9, No. 9, August 31, 1916.)—It was in 1857 that Siemens invented the regenerative furnace, but not until 1861 did he take out a patent for what we now know as the open-hearth; and even then he did not see that a new problem had arisen and that his furnace had almost miraculously appeared just when it was needed, for the Bessemer process had just come into use and large quantities of steel scrap were being produced. Numerous attempts were made to "pile" and weld this scrap, just as always has been done with wrought iron, and schemes of this kind were quite common even in the early eighties; but the welding of Bessemer scrap at that time was out of the question, because practically all the output was rail steel, while a large proportion was too large to be made into any ordinary "bundle."

For a long time after the regenerative furnace had been in operation for heating purposes it was not realized that steel scrap could be melted on an open hearth, and large piles of crop ends and defective blooms accumulated without any practicable method in sight for their utilization. But in 1864 two brothers, named Martin, operated a furnace at Sireuil, in France, and worked out the method, which is so familiar to us to-day, of melting pig iron and scrap together. They obtained patents a year later, and for a long time it was the general custom to speak of the Siemens-Martin or, more briefly, the S. M. process; but to-day we commonly refer to it as the open-hearth. These patents for many years were the subject of litigation in this country, and it was not until after 1880 that the decision was given against the patentees. Pierre Martin died at an advanced age, in an obscure French village where for many years he had lived in poverty. His last days were made comfortable by money raised by public subscription in Europe.

Self-luminous Paint. W. S. ANDREWS. (*General Electric Review*, vol. xix, No. 9, September, 1916.)—Soon after the discovery of radium by the Curies, in 1898, certain phosphorescent mineral compounds were found to be so sensitive to the radio-activity of this element that they show luminescence when only brought near it, actual contact being unnecessary. A fine quality of zinc sulphide known as Sidot's blende proved to be the most responsive. A next obvious step was to mix a very minute amount of radium salt with finely powdered zinc sulphide, combined with a suitable adhesive so that it might be used as a self-luminous paint. For a long time looked upon only as an interesting scientific curiosity, it has in recent years been applied in various useful ways to make small

objects, such as the hands of timepieces, electric switch buttons, etc., constantly visible in the dark.

This radio-active, self-luminous paint must not be confounded with an article that has been on the market for many years under the name of "Balmain's luminous paint," which was invented by Professor Balmain, of the London University, about the year 1875. The base of the Balmain paint is a special preparation of phosphorescent calcium sulphide, and it requires the excitation of a strong light to make it shine. It absorbs the luminous radiation, and thus emits it again as a soft, phosphorescent glow which gradually fades away, so that in the course of a few hours it ceases to be visible until again excited by phosphorescence. The self-luminous, radio-active paint differs entirely from the above in containing within itself its own exciting power, so that it continues to shine indefinitely, even when kept in perpetual darkness. There is, however, a practical limit to the life of the radium salt and zinc sulphide mixture, but there is no difficulty in meeting the guarantee required by the United States Government for army and navy purposes that it shall maintain undiminished luminosity for two years.

It is well known that the phenomenon of radio-activity is not confined to radium. A product of thorium, known as radio-thorium, is intensely radio-active, but it has a much shorter life than radium. Meso-thorium, from which radio-thorium is evolved, is a by-product of the incandescent gas mantle industry, and, on account of its relative cheapness as compared with radium, it is now being used extensively, either by itself or combined with radium, in the production of self-luminous paints. The relative merits of radio-thorium paint and that energized entirely by radium are unsettled, but both compounds will probably find their appropriate applications.

Problems of the Submarine. ANON. (*The Times* (London) *Engineering Supplement*. Vol. 12, No. 502, August 25, 1916.)—A difficulty in the development of the submarine is in the limitations of suitable machinery for propulsion. For surface speeds up to 18 or 20 knots, it is impossible to use Diesel engines, but unfortunately these engines cannot be constructed to give large powers per cylinder with safety. One cylinder will develop 100 horse-power quite easily and even as much as 150, but beyond the latter value little has been done. Accordingly, to obtain a reasonably large power, a great number of cylinders is required, but here again, in order not to complicate arrangements unduly, not more than twelve cylinders are usually fitted to one shaft. With a three-shaft arrangement, therefore, 5,500 horse-power is the maximum possible. Speeds of 25 knots or more are, however, desirable in submarines if they are to take their place in a squadron, and for these much larger powers are necessary. As a solution to the problem, steam machinery for surface propulsion has been projected developing as much as 15,000 horse-power. With a steam installation, special

arrangements would be necessary to avoid delay in diving, when of course the steam power would not be available. The use of oil fuel would enable the fires to be extinguished by the turning of a cock, while water-tube boilers would hold very little steam, so that the quantity of heat taken down with the boat would be reduced to a minimum. But many drawbacks must be entailed with any steam installation; one of them being that the consumption of oil is three times as great as in an oil engine, so that the radius of action is much reduced, and a decided step would be taken if oil engines could be constructed to give higher powers.

The electric drive still continues to be the only possible one for underwater use. It takes up much space and its weight is very large; about five times the corresponding figure for oil engines. On this account the submerged speeds of submarines have to be low. Further the capacity of the storage batteries limits operations submerged to relatively short runs. An escape of the battery fluid may have serious and even fatal results, and special precautions are taken to guard against such accidents. The Edison battery, though it does not save much in weight, is safer in use than the older type.

The "Lumen" as a Basis of Comparison for Illuminants. L. Gaster. (*The Illuminating Engineer* (London), vol. ix, No. 7, July, 1916.)—During recent years the practice of basing comparisons of various types of lamps and illuminating engineering calculations on the conception of the flux of light has made great progress. The lumen, originally proposed by Professor Blondel at the International Electrical Congress at Geneva in 1896, has now become a much more familiar unit to lighting engineers. The general impression at that time was undoubtedly that the expression of illuminating power in terms of lumens is the more accurate and scientific method, but that the candle-power, being now a well-understood and familiar term, could not be readily displaced. While, therefore, scientific men are turning towards the use of the lumen for accurate comparisons, in dealing with the general public an educational campaign must be undertaken in order to achieve the desired alteration gradually.

As a substitute for such terms as "candle-power per square inch" and "equivalent foot-candles" for expressing brightness, the Illuminating Engineering Society in the United States suggests as a unit "lambert, which is the brightness of a surface emitting one lumen per square centimeter of radiating surface."

A New Method of Refrigeration. W. H. EASTON. (*Scientific American*, vol. cxv, No. 11, September 2, 1916.)—The usual present practice in refrigeration is to employ an easily compressible gas, such as ammonia, carbon dioxide, or sulphur dioxide, liquefy it under pressure, and after cooling allow it to vaporize

under reduced pressure. The heat from this vaporization is absorbed from the surrounding medium, usually brine, and is used as the actual refrigerant. The newly developed system of refrigeration differs radically from this by using neither gas nor high pressure, but produces low temperatures solely through the evaporation of water. This process is a reversion to the earliest known method of cooling water, that of placing it in a porous vessel exposed to a draught of air. Only one improvement is introduced in the new system; the water is evaporated in a partial vacuum instead of at atmospheric pressure, the rate of evaporation being thus considerably increased. At a pressure of 0.0886 pounds per square inch, water boils at 32 deg. Fahr. Much lower temperatures, however, are attainable with compression machines.

A number of attempts have been made in the past to apply this well-known principle, but they have all failed through the want of a suitable air pump. The success of the new system rests on the employment of the centrifugal air pump invented several years ago by Maurice Leblanc. One of the obvious advantages of this type of installation is in the elimination of a dangerous gas carried at high pressure. On shipboard, especially war ships, this point is of great importance.

Coal Gas as a Motor Fuel. ANON. (*Motor Traction*, vol. xxiii, No. 601, September 6, 1916.)—In these times when every kind of substitute for petrol is being tested either for mixing purposes or as a whole fuel, it will be of interest to consider whether there are any commercial possibilities in the use of coal gas as a fuel for commercial vehicles. Mechanically, there is nothing to prevent the use of gas in internal combustion engines, as the alterations required would probably be only to the jet and the air supply, and the provision of a gas control valve between the gas cylinders and the carburetter; but commercially the increase of dead load which must reduce the carrying, and consequently, the earning, power of the vehicle, and a rise in such costs as tire charges, must be looked on as making the proposal unsound for the ordinary commercial vehicle.

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